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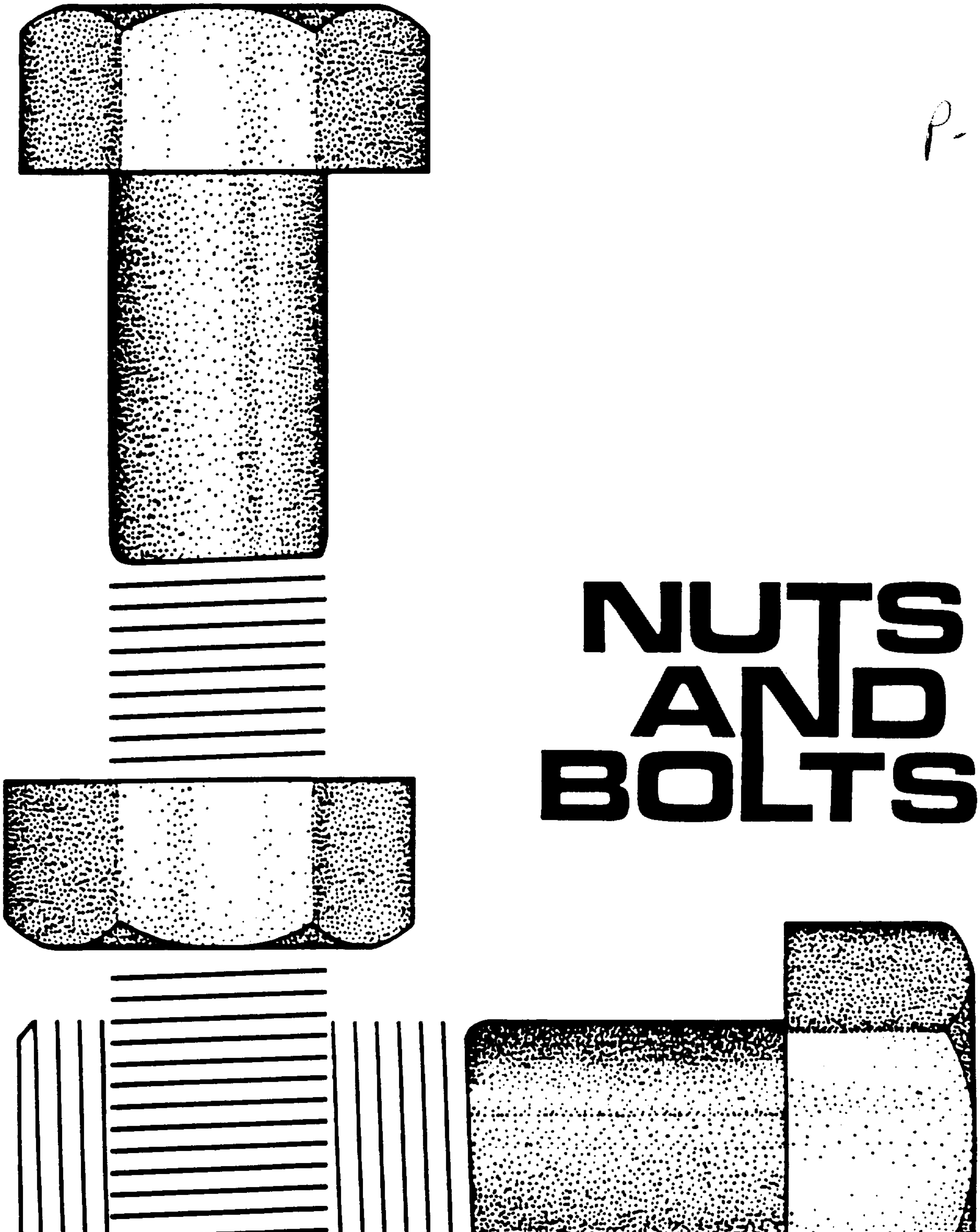
NUTS AND BOLTS



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NUTS AND BOLTS

(Practical Applications
of Bolts and Nuts
for the Designer)

by
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Code 754

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I #

Symbols Used

Symbol	Meaning	Page
A	Bolt cross-sectional area	2-13
A_b	Bolt area	4-4
A_e	Bolt deflection	3-17
A_f	Effective frame area	4-4
A_g	Area of gasket under compression	3-29
A_s	Area of steel under compression	3-29
C	Compliance = reciprocal of stiffness = $1/K$	3-28
D	Thread outside diameter	2-11
D_p'	Pitch diameter	2-11
D_w	Washer mean diameter	A-2-3
E	Young's Modulus	2-13
E_b	Young's Modulus for bolt	4-4
E_f	Young's Modulus for frame	4-4
E_g	Young's Modulus of gasket	3-28
E_s	Young's Modulus of steel	3-28
F	Applied Load	2-11
F'	Friction Force = Summation of dF'	2-7
F_B	Additional bolt load	4-4
F_E	Additional load capacity	4-5
F_e	Total added load	3-4
H_o	Horizontal reaction force	A-2-1
H_1	New screw friction torque force	A-2-1
H_2	Washer friction force	A-2-4
K	Stiffness = $F/\Delta L$	3-4
K_b	Bolt stiffness	3-4
K_f	Frame stiffness	3-4
K_g	Stiffness of gasket	3-29
K_s	Stiffness of steel	3-29

2-8

L	Bolt grip length	2-13
L_b	Grip length of bolt	4-4
L_f	Effective frame length	4-4
L_g	Length of gasket	3-28
L_s	Length of steel	3-28
N	Summation of forces dN	2-7
P	Torquing force	2-5
P	Screw advance in one turn = thread pitch per revolution	2-5
P	Applied load	2-13
PD	Pitch diameter	A-2-3
Q	Total thread friction force	A-2-1
R	Ratio of bolt stiffness to frame stiffness	3-4
S_n	Normal stress	A-3-1
S_s	Shear stress	A-3-1
S_x	Normal stress (x-direction)	A-3-1
S_y	Normal stress (y-direction)	A-3-1
T	Torque	2-11
T	Tension load	3-50
T_1	Linear dimensions of plate tear-out area	5-8
T_2	" " " " " "	5-9
V	Shear force	3-49
W	Applied vertical force	2-5
a	Moment arm of torquing force	2-5
b	Diameter of washer face	2-11
dF'	Incremental friction force	2-5
dN	Incremental normal force	2-5
e	Moment arm	3-49,50
k	Shear factor	5-3
r	Distance from screw centerline to reaction normal force line = r_1+r_2/2	2-6
r_1	Radius to inner edge of threads	2-5
r_2	Radius to outer edge of threads	2-5

3

Δ	Bolt elongation	2-13
Δ	Total deflection	3-35
ΔL	Change in length	3-35
Δ_b	Bolt deflection	3-17
Δ_f	Frame deflection	3-17
Δ_g	Gasket deflection	3-29
Δ_s	Steel retainer deflection	3-29
Δ_1	Bolt elongation at each of its ends	3-1,3
Δ_2	Frame deflection (each end)	3-1,3
$\sum$	Summation of incremental values	2-7

α	Angle of thread inclination	2-5
β	Angle between the vertical and A normal to the thread face	2-10
ε	Strain	2-13
μ	Coefficient of friction	2-1
μ_1	Thread coefficient of friction	2-11
μ_2	Washer coefficient of friction	2-11
$\sigma=\sigma_t$	Tensile stress	2-13
τ_s	Shear stress	5-8
ϕ	Angle from plane of known stress to plane of unknown stress	A-3-3

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SECTION I

PROBLEMS ENCOUNTERED WITH BOLTED CONNECTIONS

Proper joint analysis and design requires careful consideration of the materials being joined, the fasteners used, and the joint geometry. The complexity resulting from the combined effects of these variables prevents a completely accurate analysis if basic assumptions are not employed. These assumptions should be verified by joint tests, but often they are not. Some of these various aspects of joint analysis are explored by the following questions.

Q.1 Why is it so difficult to make a positive decision on what bolt and nut combinations should be used in an application, and how much torque should be applied?

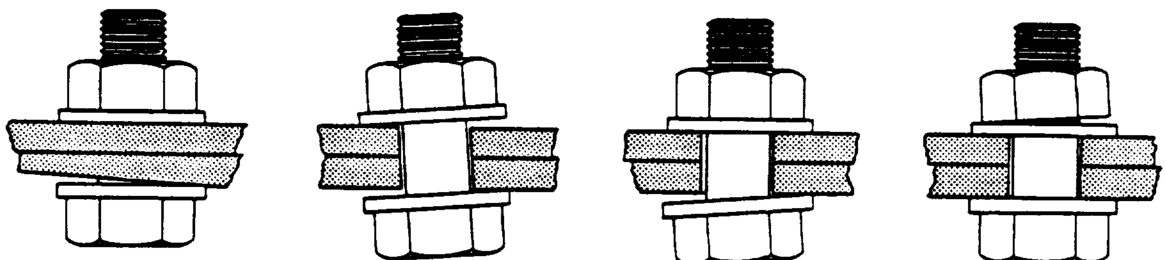
A.1 Because the root of the bolt thread and the radius under the bolt head are locations where critical stresses usually occur. It is next to impossible to measure or even calculate the stresses in these areas under loading conditions experienced by bolts.

Q.2 What other prime difficulty is most encountered with bolt selection and joint reliability?

A.2 The variations in quality control both between different manufacturers and from the same manufacturer present difficulties. Consistency is most difficult to find. Increasing costs for materials, labor and inspection have forced reduction in quality assurance by bolt manufacturers to remain competitive.

Q.3 What are some problems induced by geometry errors in bolt and nut applications?

A.3 Geometry errors can induce additional joint forces and moments as a function of geometry, as shown in figure 1.



1a — Bolted Surfaces Not Parallel

1b — Hole Drilled at an Angle

1c — Head of Bolt Not Perpendicular to Centerline of Bolt

1d — Face of Nut Not Perpendicular to Centerline of Bolt

Figure 1

Geometry errors in bolts which induce additional forces and moments

Q.4 If it were possible to determine accurately all of the forces acting on a nut/bolt joint, why would it still be difficult to predict stresses in these fasteners?

A.4 Figure 2 and the three references listed below point out clearly that there is a wide variation in stress levels along the threads, in both the bolt and the nut. Any thorough analysis of bolts and nuts must take this variation into account. The critical stress is generally in the first few threads of the bolt and in the threads of the nut next to the bearing surface.

Bolts and nuts always break at their point of maximum stress, not at "average stresses". Yet only "average stresses" are predicted by most analyses.

If a bolt were tightened up to its yield point (and some engineers would define the yield point by an analysis which predicts only the "average yield point"), then there would be local points in bolt threads which would already have yielded. These local points driven into their plastic yield regions would not spring back to their original positions when the bolt load is released. This is one major reason why bolts which have been torqued up to a certain value a second time will not produce the same clamping force.

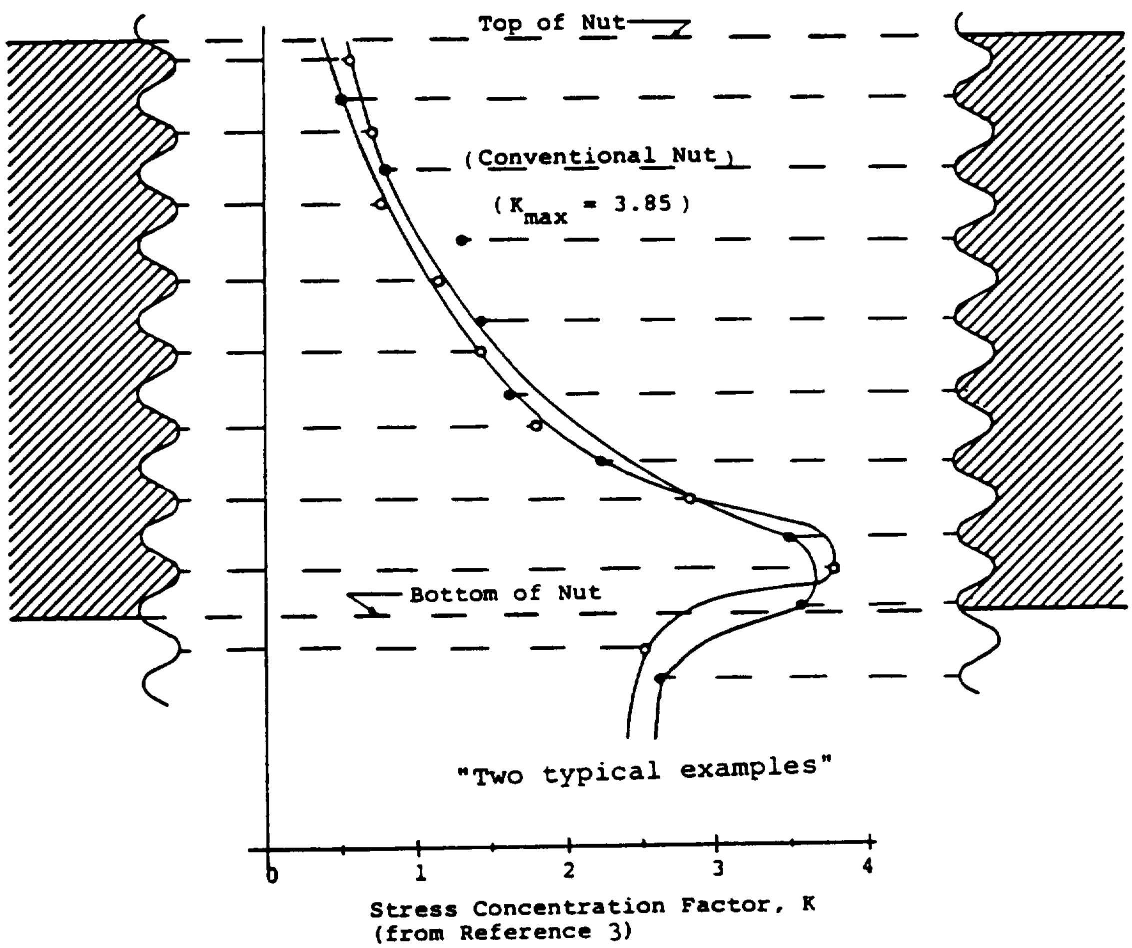


Figure 2a - Theoretical Stress Distribution in Nuts and Bolts

References:

Kulju, Ken, "How to Design High-Strength Bolted Joints," Machine Design. May 1967. (3)

Cnalupnik, James D., "Stress Concentration in Bolt-Thread Roots," Experimental Mechanics. September 1968. (4)

"Controlling Fastening Reliability and Costs," (Editorial) Assembly Engineering. January 1973. (5)

Fabrication methods for bolt heads and threads will determine the crucial fillet stress concentrations around the heads and in the threads. Controlled head forgings, for example, will form uniform grain flow with unbroken flow lines. This unbroken grain flow reduces stress concentrations

The thread formation and its "runout" are most critical to high stress concentrations. Machined "V" shaped threads soon develop cracks compared to smooth radiused roots in formed or rolled threads.

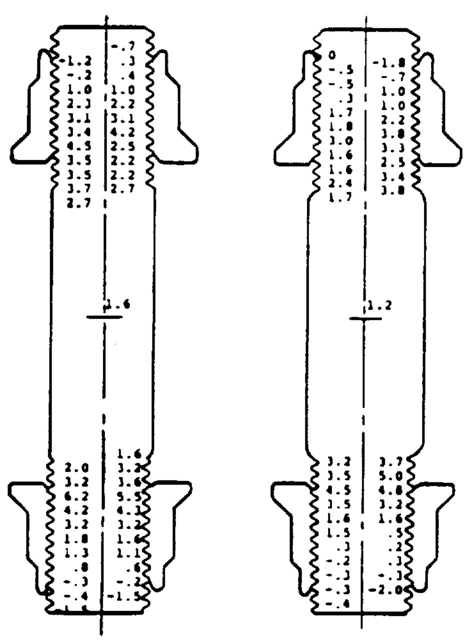


Figure 2b - Fringe order maxima for standard and oversize bolts. (Standard nuts are at bottom; high numbers give high stresses.) (From Reference 4)

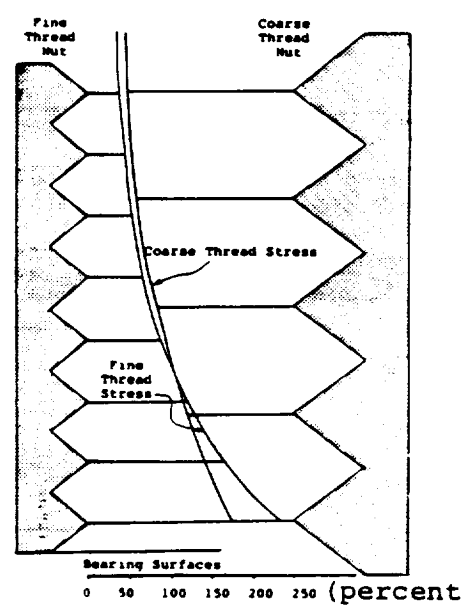


Figure 2c - Strength advantage of coarse threads in nuts is due to more uniform load distribution in coarse threads (from Reference 5)

8

Q.5 Why are so many difficulties presented before any solutions are offered in this manual?

A.5 It is a good approach to consider all of the important difficulties before offering solutions. It is better to be aware of these design problems than to begin solutions using assumptions which may be either misleading or wrong. In addition, there are many design techniques which eliminate most of the difficulties presented. However, it is impossible to get around a design difficulty if it is not considered.

Q.6 What is one example of an important joint geometric variable which the designer can circumvent by a proper design?

A.6 One geometric variable which has a profound effect on the structural integrity of a joint and its fatigue life is nut angularity (as illustrated in figure 3).

Some conclusions have been developed from results of fatigue tests with various inclinations of nut seating (see Reference 7).

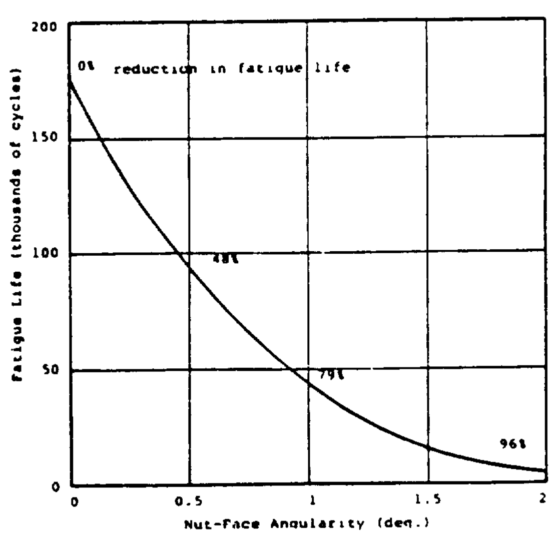


Figure 3 - Effect of Nut Angularity on Fatigue Life (See Reference 6)

9

Inclination of the nut seating will have an adverse effect on fatigue endurance and strength, the effect being more pronounced at long endurances. For example, a one-degree inclination can reduce fatigue strength at 10^7 cycles by 40%. Additional discussions are given in the following references.

Kiddle, F. E., "Variation of Bolt Fatigue Life with Inclination of Nut Seating," Royal Aircraft Establishment, Technical Report 67174. July 1968. (6)

"Nomogram for Bending Stresses in Bolts," reprinted from Aug. 21, 1972 issue of Design News, a Cahners Publication.(7)

In Reference 7 above, a nomogram for bending stresses in bolts is provided. This article gives direct stress calculations for the amount of stress in a bolt due to the angularity induced bending, which can be determined by measuring the gap under the head.

This one variable has an adverse effect on both bolts and nuts. However, the designer can circumvent this problem by taking proper design precautions, such as the following:

- Make sure that the plates which are being bolted together have flat interface surfaces.
- Make sure that the head of the bolt and the nut meet controlled flatness requirements.
- If the surfaces cannot be made flat, use angle washers to bring the surfaces to the required flatness.

Now it has been shown that two of the most common bolt design assumptions can lead to joint problems (i.e., that all bolt threads take the load equally, and that bending stress in bolts due to seating surfaces and bolt manufacturing are negligible). These design assumptions are not minor assumptions. On the contrary, these two factors contribute considerably to the actual critical stresses in both nuts and bolts.

Q.7 How many other major difficulties should be investigated before a designer can consider the direct "average tensile stresses" in a bolt-nut combination?

A.7 A list of some of the major problem areas in bolt design is given below. (Some practical methods for elimination of these problem areas or ways to get around them in design or by methods of testing and analysis are given in subsequent sections of this manual.)

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1. • The stress distribution across the threads may not be uniform.

2. • A nut may be manufactured with its interface surface not perpendicular to the axis of the bolt hole.

3. • The bolt head bearing surface may not be perpendicular to the axis of the bolt hole.

4. • The axis of a bolt hole may not be straight.

5. • Part mating surfaces may not be parallel when bolted together.

6. • Lubricating the threads and under the head, may cause stress variations in the bolt.

7. • Types of materials used for bolts and nuts may not be suitable or compatible.

8. • The variables and incompatibilities in the materials clamped together, including the washer materials may present problems.

9. • The methods of quality control in manufacturing may be poor.

10. • Analytical methods used to analyze "average loads" and "average stresses" in bolts may be suspect compared with actual peak stresses occurring within the bolts.

11. • Methods of part inspection and quality control may be poor on a project.

12. • Methods of torque application may not be reliable and may induce additional stresses.

13. • Vibration and shock applications for bolts and nuts could present problems.

14. • Methods used to prevent shock and vibration may be inadequate.

15. • Problems may be encountered when using the bolt as a stud (without a nut), including the use of inserts.

Q.8 What is one practical way to circumvent many of the above variables?

A.8 There is one simple experimental method which can be used to verify the torques and preloads applied to a joint. Figure 4 illustrates this method in which two plates are held together with a bolt and nut combination. This method requires prototype nuts, bolts and washers to perform a simple test on a prototype system joint. If lubrication is to be used in the assembly,* lubricate the threads and under the head of the bolt and nut (to keep the friction variables reduced as much as possible). Make sure that the faces of the plates are smooth and flat. Use only bolts, nuts or washers from reputable manufacturers which can be trusted for their quality control and consistency. Assemble the prototype bolt, nut and washers on the plates so that the nut can be drawn up hand tight. Measure the length of the bolt and the thickness of the plates with a micrometer. Take a quality calibrated torque wrench and turn the nut up to 10% of its computed torque to reach yield (see manufacturers bolt torque tables or torque estimates given in section 2 below). Hold the bolt from twisting with a second wrench. Mark the plate as indicated in figure 4. Then apply 30% of the yield torque and mark the plate again. Repeat the same processes for torques of 50%, 70% and 90% of the estimated yield torque. If the yield torques were calculated properly, the angle of turn would be appreciably higher than the others when the 110% of yield torque was applied. If the theoretical torque to yield was calculated too low, then the increment from 90% to 110% would have been the same as the other increments. Continue the torque and mark procedure until the yield point is reached. Then re-measure the bolt length. This yield torque will be an accurate measurement of the combination joint tested with all of its inherent variables. Note, however, that the yield actually reached could be the yield of either the bolt, the nut, or even the plates being compressed. However, it is most important to know only the

*The use of lubricated threads may not be permissible in some applications such as space or vacuum systems. Clean, dry, degreased systems should then be used.

[redacted] 12 N

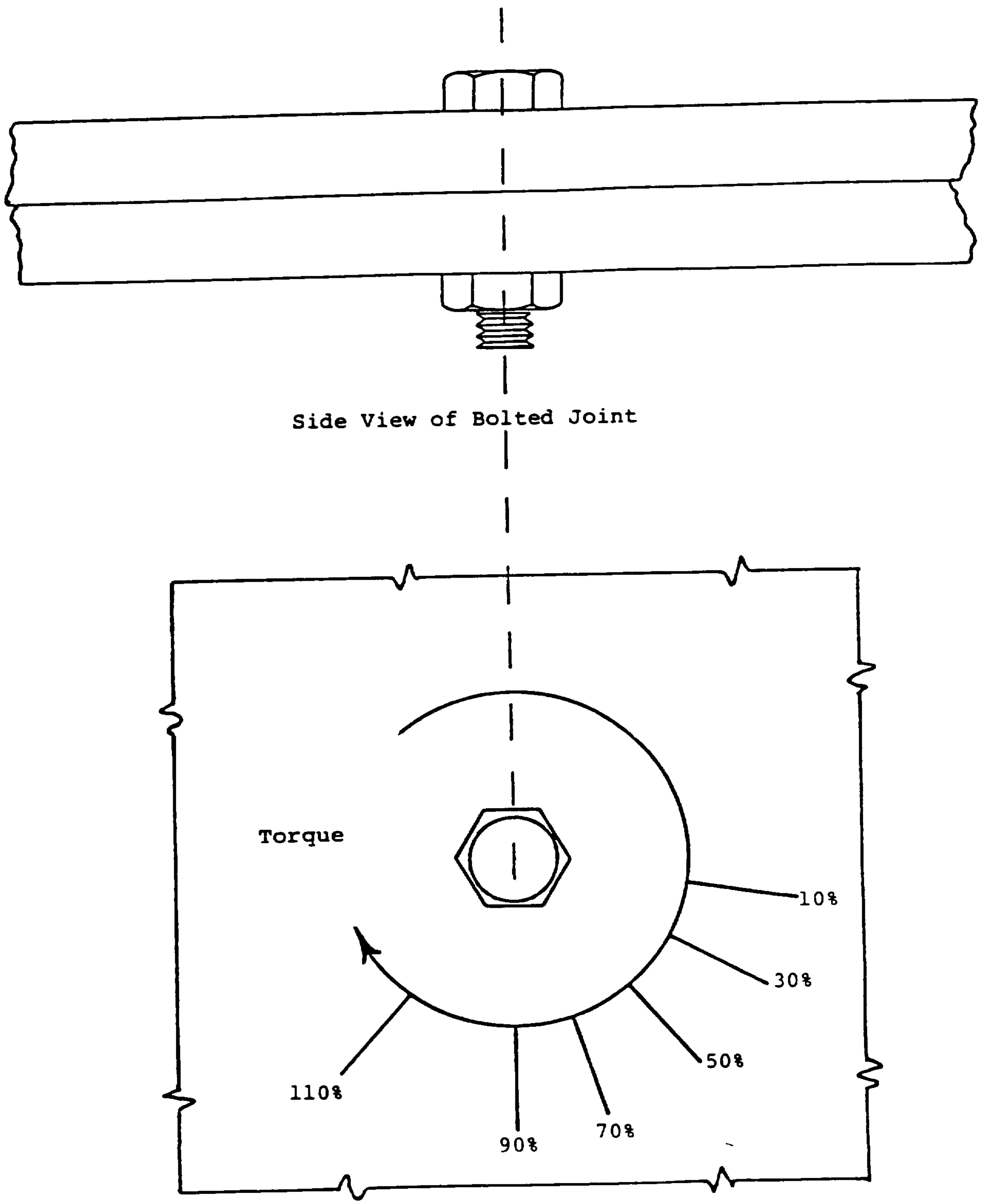


Figure 4 - Torques Applied to a Bolt and Nut Which Hold Two Plates Together

yield of the joint combination to make a proper design. Repeat the above procedure for several bolt and nut combinations to insure reliability. If the yield torque is reached before 100% of the expected torque is applied, then adjust the combination design to consider the new yield point found. The approximate average stress in the bolt at any torque is found to be equal to:

$$\text{Average stress} = \frac{\text{Young's modulus for bolt x change in bolt length}}{\text{original bolt grip length}} \qquad (1)$$

In the final joint design, use a torque value equal to only 60% of the actual yield torque experimentally determined. Use this torque value for the rest of the bolts in a design. Half of the unused torque capacity will allow for increased stress due to variations in manufacturing. The rest of the unused torque capacity will allow for the additional bolt loads that will occur in these bolts during operation.

Return now to the previous list of difficulties (page 1-7) and note how many of the problems were circumvented.

It was not necessary to measure the actual stress distribution in the threads. The torque causing yield of the assembly was simply reduced by a safety factor to take care of the unknown stress concentration factors to arrive at a final design torque.

The second, third, fourth and fifth problems listed (i.e., geometric problems and quality control problems) were controlled by using good plates and quality controlled bolts and nuts. By conducting a number of tests, the variations in these parts will be kept to a minimum. If during the tests a large torque variation between tests had been found, an investigation would have to be conducted to find out where the quality control in the parts failed. One can expect good bolt-nut torque consistency with this test combination which is, of course, the primary purpose of this test. One cannot expect exactness in an assembly where the individual parts do not have consistency in their production.

To take care of the sixth problem area, good results can be obtained by lubricating the threads and under the head of the nut and bolt (only if permitted in the final design). When the proper lubricant material is used and the personnel become skilled in this experimental technique, then the results can be consistent.

Problem area seven could be controlled by using high-strength alloy bolts (i.e., 8630 carbon steel or similar materials for bolts) which are much more consistent than low carbon bolts. Experience in the use of many types of materials will show the accuracy that can be expected from that material.

To overcome problems eight and nine, tight manufacturing reliability and quality control are a must. However, remember that the user must always provide the final quality assurance inspection. A sample bolt specification and method for bolt inspection will be presented later in this report as a guide.

It has been previously pointed out that rolled or formed threads are much more reliable than cut or ground threads. Some manufacturers do not maintain a consistent manufacturing method for threads and tests, and inspections should be constantly performed to insure this reliability.

The steel or other alloy materials used should be certified by tests for each lot of steel used in the manufacture of the bolts. A metallurgist should also certify the results of these tests. Other tests could be certified with careful quality control. Good bolt manufacturers will always supply this information with each lot of bolts upon request.

In the experimental procedure described above, problem area ten was circumvented by determining the <u>exact</u> yield torque (by measuring it directly).

To overcome problem area eleven, quality control inspection methods will be discussed in a later chapter. However, this test by itself, if consistent, will supply the quality control necessary for most design cases. The method presented here is also simple and inexpensive. It further is easily mastered by the average person. Some alternate methods will be discussed below.

Problems thirteen and fourteen, which relate to shock and vibration conditions, will also be discussed in later sections. For the average bolt-nut combination however, the best assurance against shock damage and vibration failure is a tight bolt.

Q.9 For high-strength bolt and nut applications, what combination or ratio of bolt to nut strengths would give the most reliable performance under severe shock and vibration loads?

A.9 The following sketches illustrate the areas of peak stresses in both the bolt and nut, and the most likely modes of failure.

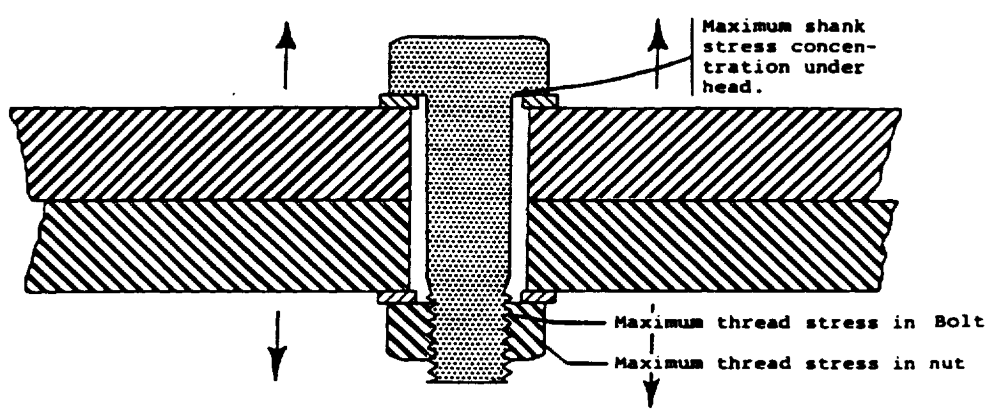
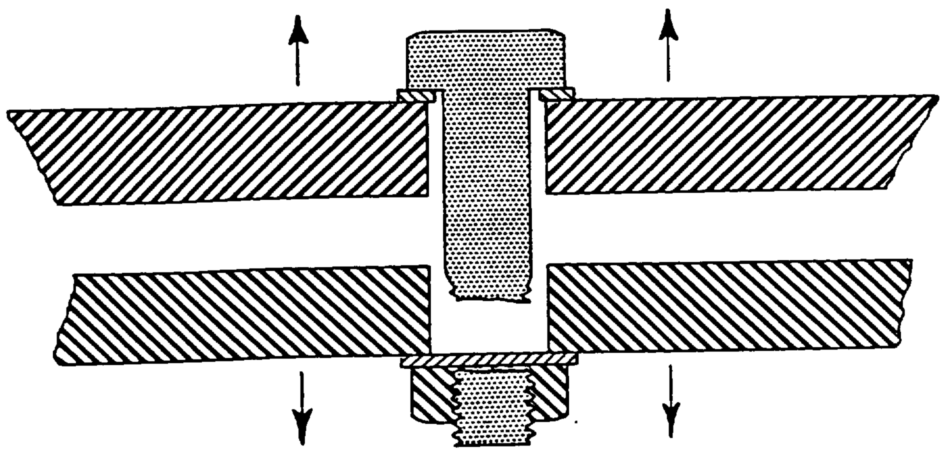
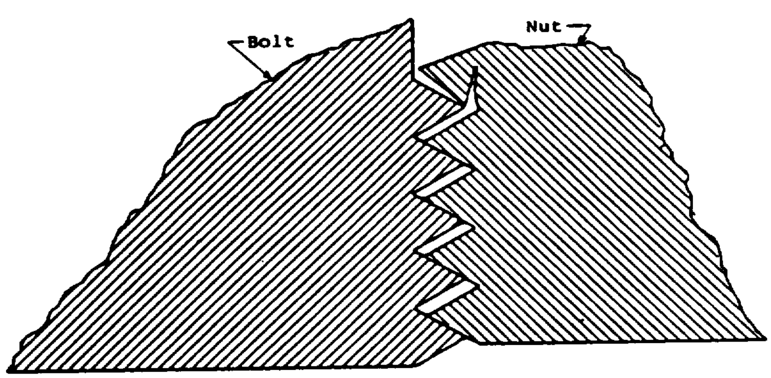


Figure 5 - Location of Maximum Thread Stress in Bolt and in Nut



Failure usually occurs adjacent to the first or second thread of a bolt. This failure usually results in a catastrophic failure of the bolt-nut assembly.

Figure 6a - Typical Bolt Failure



Failure of first or second thread in nut is usually not catastrophic as the rest of the threads of the nut are ready to take up the load. Some nuts have the first few threads removed or cut back to re-distribute the load to other threads more uniformly.

Figure 6b - Nut Thread Failure

Figure 5 shows the critical stress areas in both the bolt and nut in a configuration where a bolt and nut are joining two plates under a tensile load. After these plates are bolted together, it can be assumed that they are pulled apart by this tensile force. The critical stress location in the bolt is found at the top or uppermost threads in figure 6a.

A bolt generally will fail close to this typical critical stress area. Since there are usually too few engaged threads left above this point to hold the load, a catastrophic failure results.

On the other hand, the nut has its critical thread stresses at the head of the nut nearest to the plate or bolted surface interface (as in figure 6b)*. If the first thread in the nut fails, there are usually other threads ready to take up the load. Figures 2a, 2b and 2c demonstrated that the bolt stress level was much higher within the first few threads and dropped off to a low value at the opposite end of the nut. Additional loads could be added to these outside threads without failure. This is particularly significant if a shock load or a one-time heavy load is exerted on the assembly. This excess nut load capacity is often used in nut designs.

Since it is to the advantage of the assembly design to allow the first several threads of the nut to yield without breaking, it would be of advantage to use a material in the nut which has less strength and a higher yield elongation than the bolt. (Elongation is the change in length per unit initial length.) Thus, a good combination would be a strong hard bolt of 180,000 psi strength with a limited yield elongation, and a nut with 120,000 psi allowable stress with a greater yield elongation. Unfortunately, most high-strength bolts manufactured today have low elongations. Current military specifications require only a 5% elongation to bring a bolt to its yield point. If bolts

*METALS HANDBOOK - Vol 11 - NONDESTRUCTIVE INSPECTION AND QUALITY CONTROL, ASME, Metals Park, Ohio 44073.

were designed to this specification, they would generally take very little additional yielding before failure occurs. To avoid this condition, it would be best to simply specify the bolt with, for example, 180,000 psi material and a nut with 120,000 psi material. Greater total elongation can also be achieved by using longer nuts.

It is a further advantage to specify that the threads of the bolt and nut be a close fit (i.e., a class 3A and 3B) to obtain better load distribution in the threads. In this way, as higher loads are applied, the nut threads will yield to fit the threads in the stronger bolt. Although the threads in the nut may yield, they generally would not break. They would therefore pass more of the load on to the next threads in the nut. This type of yielding may destroy the nut for further use, but nuts are much less expensive than bolts, and catastrophic failures would be kept to a minimum by this material selection technique. Practical design examples of this technique using quality controlled one-inch diameter bolts and nuts have demonstrated that although bolts may be used as many as 25 times, the nuts must be discarded after only three uses.

A word of caution must be given for those designers who would desire to use this philosophy of making the tensile strength of the nut lower than that of the bolt. If the yield stress in the nut is made extremely low, the nut could split open or it may yield too far and simply allow the nut threads to slip past the threads on the bolt. This last effect is compounded by the fact that the root angle of the threads in both the bolt and nut produce radial outward force components which are supported in the nut by hoop tension.

Q.10 Why have some designers recommended a highly heat treated or hard nut material (i.e., a hardened nut, say 180,000 psi yield stress) along with a bolt of lower strength (say 160,000 psi)?

A.10
- Some designers believe that an over-torque would only snap the nut off (or break it in hoop tension). They call this a safety factor or a "fuse" protection against over-torquing.
- Some designers expect that a hardened nut is more linear than a nut of 120,000 psi and believe that it would therefore be less likely to yield. Thus, such a nut is assumed by them to be more reliable and can be used over and over.

The rebuttal to the first point is that there are recorded cases of hardened nuts which have been cracked during assembly torquing and which were not noticed until the load had been later applied.

The response to the second point is simply that local yielding exists in almost all bolt-nut applications when the nut is torqued up to a point just below its yield. Recall also that the torque wrench only indicates an "average" yield torque. Recall also that the yield stress in the threads as graphically illustrated in Figures 2a, 2b and 2c would imply that some points may have stresses higher than their yield stress.

Q.11 If the stresses in the threads are so critical and so indeterminate, why are there not many more failures?

A.11
- There are many bolt/nut failures. Many of these failures have caused tragic results. Other failures are simply not recorded. It would be interesting to count the number of lug nuts turned off car wheels in the United States and to record the types of failures observed on these lugs.

  Some local joint failures are not recorded since they are easily passed over during inspection or are not recognized. With current liability laws, there is a tendency simply to correct a problem and not attempt to isolate its cause.

- Most commercial bolts are made of materials which have their allowable stresses in a low stress range (i.e., 40,000 psi to 60,000 psi). Thus many of them could yield locally and wouldn't completely break. From this type of local yielding failure, the following action often occurs: the load is passed off to another bolt or another part which is subsequently damaged; threads are damaged or distorted which makes the bolt/nut combination almost impossible to take apart (this problem is often experienced by automotive and appliance repairmen); yielding of the thread could cause the bolt and nut combination to lose its preload. This loss of preload would make the assembly susceptible to failure under any shock or vibration loading condition. Local failures can also lead to jamming of parts.

- Since many bolted joints support their loads in shear, the additional tensile load due to a local thread failure could be minimal. Therefore, while problems exist, they may not always be critical.

- Because of the above difficulties, many designers have over-designed bolted joints. This practice could also lead to a different type of failure, due to the loosening of a series of stiff bolts. This could occur uniformly in a bolt pattern or sequentially. A sequential loosening of a set of bolts could cause the last bolt in an assembly to fail.

- In some cases the design load predicted is lower than the actual field experienced load but the factor of safety used is quite high. For example, in most hoist and crane problems, the load factor used is six. Factors as high as this would allow for many errors in bolt design. Spacecraft designs are at the other extreme, and use load factors as low as 1.25.

- In an article published in "Assembly Engineering"[8] (December 1970, p. 24), it was stated that American Airlines estimates that more than 10,000 man-hours per year are required to drill out seized fasteners. This is an obvious example of the large numbers of bolt failures. These are perhaps not catastrophic, but they are expensive.

Q.12 As this brief look at bolt and nut problems is finished, and an examination of more detailed design problems is started, what are some of the practical detailed joint design questions with which a designer should be concerned?

A.12 Leading a check list of the specific questions which the the designer should be concerned with are the following:

- How much actual torque should be specified for a joint design?
- Should a lubricant be specified for the threads of a joint? If so, how much?
- Should the project go to the expense of using high-strength bolts?
- Should the project design use hard or soft washers, and why?
- Should the design use lock-nuts or ordinary nuts? Should it use lock washers?
- Should a machinist prepare the surfaces before bolting the assembly together? How much surface preparation is required?
- Should the design demand ductility in the bolts or nuts?
- What plating materials should be used on the bolts? What plating materials should be used on the nuts?
- What are some of the effects of improperly applied surface finishes and electroplating?
- Will temperatures have an effect on the bolted assembly?
- What kind of safety factor should be used for the design loads and for the materials?
- What kind of safety factor should be used on the expected bolt and nut load capacities themselves?
- Do dynamic loads affect the assembly? What considerations should be given to the design to anticipate shock and vibration loads?
- Has the fatigue life of the bolt and nut been properly accounted for in the design to meet the life requirements of the entire assembly?
- How can bolt specifications be written to assure that the design incorporates what is really needed with the expected systems reliability and without incurring excessive costs?
- Is there a way to check the actual metallurgy of the hardware being used?
- Are all bolts marked in such a way that mechanics using them are always aware of the type of bolt or nut they are working with?

- What are the effects of over-yielding in the bolt during the tightening operation? What are the effects of yielding it more than once?
- What installation tools are most reliable in the torquing operation?
- How reliable is the "turn-of-the-nut" method?
- How tamper-proof are the bolts and nuts after they have been torqued? (Not necessarily by vandals, but with well-meaning mechanics who give the nut an extra twist "just to be safe".)
- Even though the exact stress distributions in the threads and under the bolt head are not known, what can be done to keep these stresses to a minimum?
- What effect does the manufacturing process have on the performance of bolts and nuts?
- What are some of the commercially and military established design criteria for bolted joints? How do these different criteria compare?
- What materials should be used with caution in a bolted joint combination?
- How should gaskets be tightened to prevent their flanges or retainer covers from being damaged?

The designer should always make a similar check list, review it, and attempt to answer each question before he is satisfied that the bolted joints he has configured will have adequate integrity for their intended job.

In the following sections, some specific approaches will be outlined for developing answers to some of these practical design questions.

Q.18 What are some classical relations between applied torques and bolt tensions?

A.18 The report, "The Relation between Torque and Tension for High Strength Threaded Fasteners," by J. I. Price and D. K. Traask, U.S. Department of Commerce Bureau of Standards,[14] is an excellent report.

- The relations developed in this reference between applied torques and tensions are generally valid for torques applied to the bolt head or to the nut. (It is assumed that the bolt is relatively short and lubricated on its threads.)
- The length of the bolt does not have an appreciable effect on the torque-tension relations for high strength fasteners tested (except for extremely long bolts).
- Self-locking devices influence the torque-tension relations. Repeated installations of a fastener through the same locking device causes a lessening of the locking torque. We shall explore this development in more detail below.

Q.19 What are other references describing the effects of lubrication on bolt torques?

A.19 Several additional references include "Torquing Stresses in Lubricated Bolts" by Roland L. Roehrich, Westinghouse Electric Corporation,[12] and "Tightening Threaded Fasteners" by Bengt Westerlund in Assembly Engineering, June 1977.[13] In this last reference it can be found that the reaction force components due to the applied torque can be assumed to be distributed into the proportions shown in figure 8.

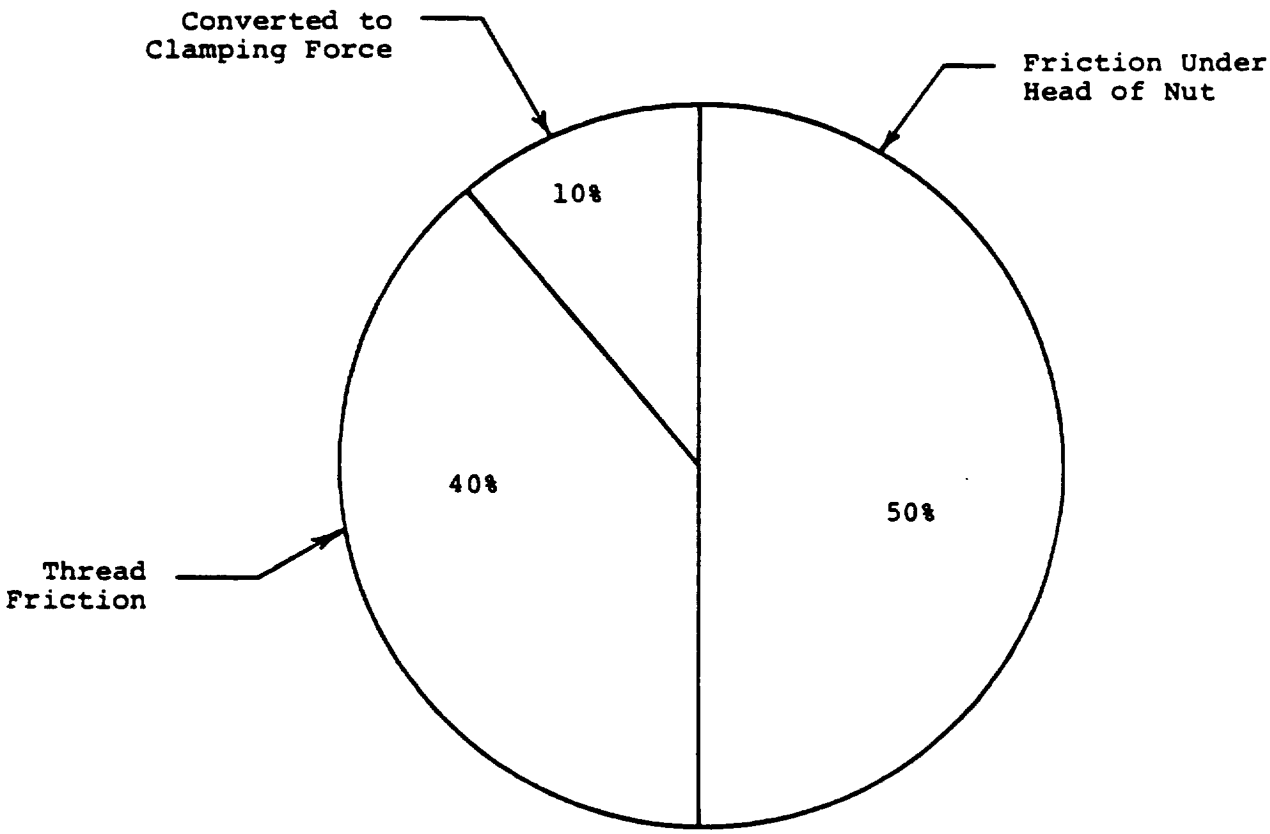


Figure 8 - Torque Reaction Forces Breakdown

Q.20 After friction, what is the next important variable to be considered in the torque/preload relation?

A.20 The next logical variable should consider the force balance acting on the bolt during its torque-up. To do this first consider a square thread which is easy to understand.

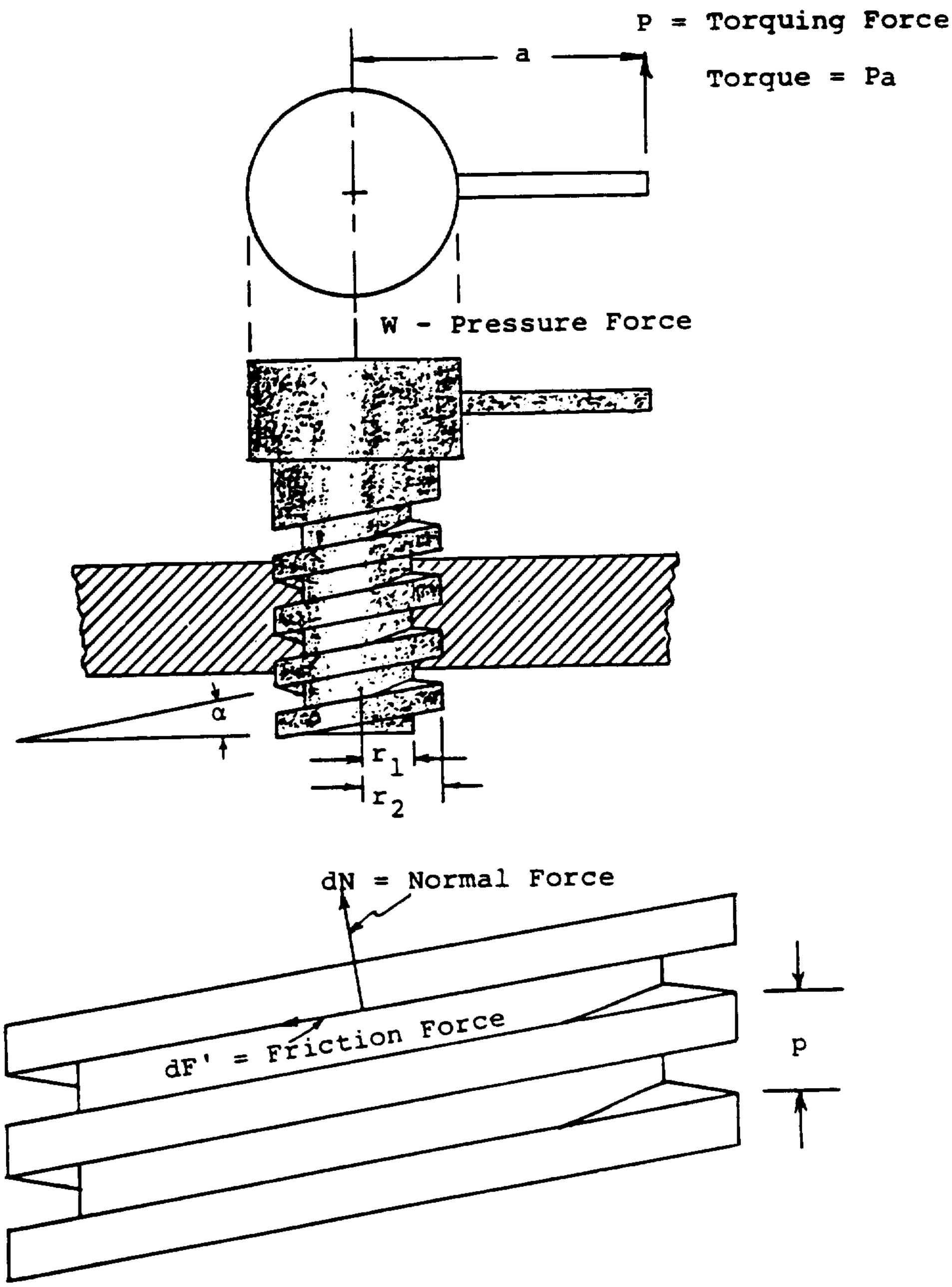


Figure 9 - Force Balance on Bolt

26N [redacted]

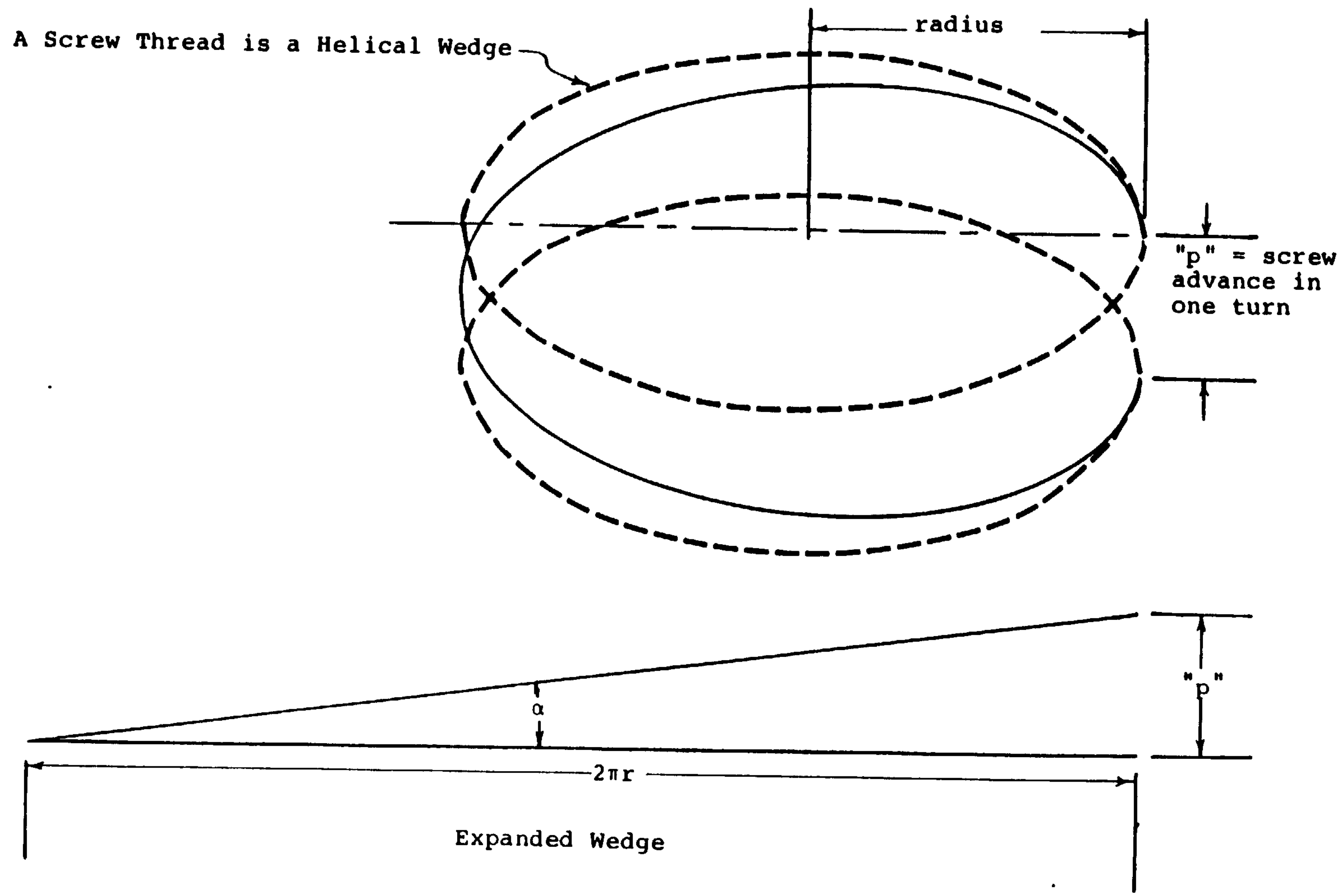


Figure 10 - Screw Thread as a Helical Wedge

Refer to Figures 9 and 10 which show a helical square threaded jack-screw. A torque equal to P.a* is applied to the jack to lift a load W.

Q.21 What is the angle α?

A.21 The incline angle α is calculated using a helical wedge of height "p" with a base of length $2\pi r$, where r is the distance from the screw centerline to the reaction normal force line, often taken as the mean radius $r = (r_1+r_2)/2$, and "p" is the thread pitch.

Since α is generally small, then $\tan \alpha \cong \alpha$ or $\alpha \cong p/2\pi r$ (in radians).

Q.22 What are the forces which act on a single thread?

A.22 Referring to figures 9 and 11, the forces acting on a single thread are explained below.

- Let "W" be the total weight to be lifted or the vertical clamping force acting down along the bolt axis.
- "N" is the summation of forces dN or "$\sum dN$"+ for all threads and acts opposite to "W".
- "F" is the friction force which is a result of "W" and acts to resist the applied torque Pa.
- Note also that the reaction torque is always opposed to the applied motion.
- F', "N", and "W" all have components in the vertical and horizontal directions and they must therefore balance.
- The forces which produce torques are always perpendicular to the vertical axis of the bolt. The applied torque is also horizontal.

*P.a means the force "P" multiplied by the moment arm distance a.

+The symbol $\sum$ is used to represent the sum of small incremental values.

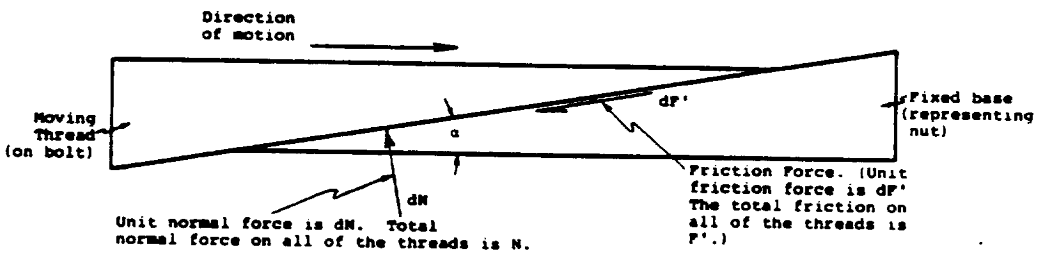


Figure 11 - Friction Forces acting on Thread

Q.23 What is the relationship between the applied torque Pa, the coefficient of friction μ, the mean radius of the screw thread r, and the applied axial force W?

A.23 In Appendix A1 of this manual, the following formula is developed which relates these variables.

$$Pa = Wr \left(\frac{\tan \alpha + \mu}{1 - \mu \tan \alpha} \right) \tag{2}$$

where $\alpha = \dfrac{\text{thread pitch}}{2\pi r}$

This result can be summarized as follows:

Given a jack screw with square threads as in figure 9:

- o If the screw advance or its pitch "p" per revolution is given, then α can be calculated. If the coefficient of friction "μ" is known, the mean radius of the threads "r" is known, and if the weight "W" to be lifted (or clamping force) is also known, then the necessary torque Pa can be calculated from formula (2) above.
- o If the screw advance "p" is given, the coefficient of friction "μ" is known, the mean radius of the threads "r" is known. The maximum weight or clamping force "W" available for a given torque Pa can be calculated.
- o Given the "W" to be lifted, and given the available torque "Pa" and if α and μ are also given, then the screw radius "r" can be calculated.
- o Given the weight "W" to be lifted, given the available torque "Pa", given the screw radius "r", given the friction coefficient, then the advance of one thread, which depends on α can be designed.

It should be emphasized, as in the case of high strength bolts and nuts, that rolled or formed threads are desirable, and in many specifications , formed threads are mandatory even when the design problem consists only of selecting the proper thread such as a 10-32 or ¼-20 thread. In most cases the same can be said of a jack screw thread where the sizes are generally available from catalogues.

Another factor to remember is that although a vast majority of bolts and nuts can be supplied from off-the-shelf stock, there are six variables in the above equation, allowing many variations in the selection. There are also other overriding variables which may not be included, such as:

- Although a single larger bolt usually gives a better lead angle (α), it is usually desirable to have several smaller bolts in a design to take the load, as several bolts can always take local bending in a connecting plate. If only one bolt holds the structure together, and it fails, there usually is a catastrophic loss.
- The Coulomb coefficient of friction (μ) is usually quality controlled to assure a reliable torque reading and uniform bolt loads.

This method of torque and preload analysis developed above will be continued and a new variable, i.e., the tapered thread instead of the square thread previously used will be introduced.

Q.24 What methods of analysis are currently available for standard threads?

A.24 One classical reference which predicts torque and preload relations is: Analysis of Nut and Bolt Torques, by James E. Foisy R64SE45 Class 1 July 31, 1964, General Electric Corp. Report.[15]

Q.25 What is an added variable in this analysis which was not considered in square threads?

A.25 The angle β which is the angle between the vertical and a normal to the thread face. (See Figure 12.)

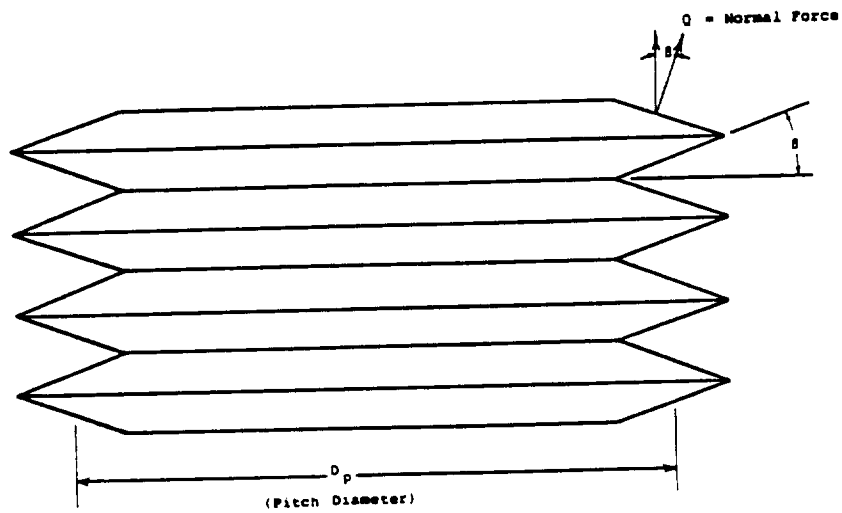


Figure 12 - Tapered Thread with Angle β

Q.2 6 How are all of the variables related to this variable and to the torque load relation analysis?

A.2 6 The torque T is related to the applied load F, and to the pitch diameter D_p as follows (from Appendix A2).

$$T = F \frac{D_p}{2} \left\{ \operatorname{Tan} \alpha + \mu_1 \operatorname{Sec} \quad + \mu_2 \frac{(D+b)}{2 D_p} \right\} \qquad (3)$$

where D = thread outside diameter

α = thread lead angle

β = thread form angle

b = washer face diameter

μ_1 = thread coefficient of friction

μ_2 = washer coefficient of friction

3a [redacted]

Q.27 How is this equation made practical today?

A.27 Nomographs, slide rules and computer programs are used to solve bolt, nut and stud torque load relations. Further, equation (2) can be modified as follows: (a) to optimize bolt size, (b) to obtain desirable lubrication friction coefficients (for most effective friction), (c) Variables can be optimized to give the most reliable torque values. (d) For machined threads the screw design can be varied to give the most desirable lead, "D_p" or the best thread angle β. (e) To determine the optimum washer conditions, such as desired lubrication (controlled by μ_2), stiffness and hardness to eliminate galling, etc.

Q. 28 Give some specific references where this equation is utilized in design problems?

A. 28 Machine Design, Hall, Holowenko, Laughlin, Schaum Publishing Co., New York, 1961, p. 146.[16]

Torquing Stresses in Lubricated Bolts, Roland L. Roehrich, "Machine Design", June 8, 1967, p. 171. [12]

Mechanics for Engineers, Statics and Dynamics, Charles O. Harris, General Motors Institute, The Ronald Press Co., N.Y.

Nomogram for Torque on Bolts, "Design News", May 22, 1972.[17]

Fastener Tension Control - "What It's All About", Terrence Thompson, Assembly Engineering, November 1976, p. 22.[18]

TURN-OF-THE-NUT-METHOD

Q.29 What is the turn-of-the-nut-method?

A.29 This is a preload method which can be described by the following bolt behavior. Because of their elastic modulus, steel bolts are stretched approximately .001 inches for each inch of grip length when the stress in the bolt is approximately 30,000 psi. (See Figure 13.)

[redacted] 33

Example bolt load

$E = 30 \times 10^6$ psi

$L = 1$ in

$P = 30{,}000$ lbs

$A = 1$ sq in

$$\frac{(30{,}000)(1)}{(1)(30{,}000{,}000)} = \Delta = .001 \text{ in} = \frac{PL}{AE}$$

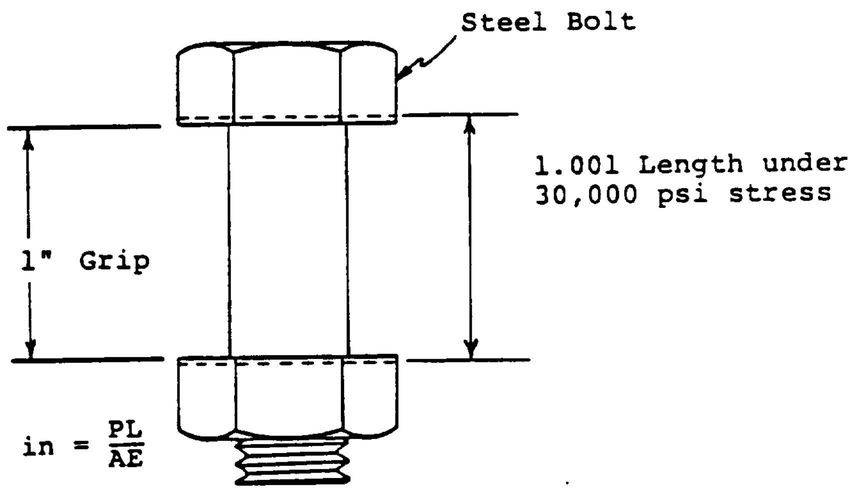


Figure 13 - Turn-of-the-Nut-Method

Since the tensile stress in a bolt is (from equation 1, page 1-10) $\sigma = P/A = E\,\Delta/L$ where E is Young's modulus, Δ is the elongation and L is the bolt grip length, this deflection Δ is independent of the area of the bolt. The equation (1) can be rewritten as $\sigma L/E = \Delta$. Therefore if the desired preload stress σ is 30,000 psi and E is 30,000,000, the equation gives

$$\Delta = \frac{30{,}000}{30{,}000{,}000} L$$

or, solving for the strain, i.e., (Δ/L = strain = ε) gives $\Delta/L = .001$; which simply means that when any steel bolt is stressed up to 30,000 psi, there would be a stretch of .001 inches per inch of grip (independent of the diameter of the bolt). A two inch long bolt grip for example will stretch .002 inches per 30,000 psi. If a two inch grip bolt is to be pre-stressed to 90,000 psi, the extension will be .002 times (90,000/30,000) = .006 inches.

A quarter inch 20 bolt means that the bolt has 20 threads per inch or 1/20 = .05 inches pitch per thread. Therefore, every time that this bolt (i.e., 1/4-20) rotates one turn, it advances .05 inches. For example, if a four inch grip 1/4-20 bolt is to be prestressed to 90,000 psi, it will have to be stretched .012 inches. If we assume that all of the applied work goes into stretching the bolt only, and none of the work goes into compressing the frame or plate material being bolted (which is not possible), then the bolt would have to be twisted .012/.05 = .24 turns beyond its initial seating position.

If the bolted material compresses the same distance that the bolt extends, the bolt will have to be turned an additional amount (i.e., 2 times .24 = .48 turns or 1/2 a turn).

Q.30 This sounds like a rather simple straightforward computation. Why is this method not used more often?

A.30 Refer back to our previous section on bolt torques. The ratio of the amount of tension the bolt experiences during pre-torque and the amount the frame compresses during pre-torque is not a fixed quantity. It depends on the effective plate or frame area under compression. This value is affected by the use of washers, etc.

Another critical variable is the flatness of the plates being bolted. The nut has to be turned up snug before measuring the turn of the nut. For example, two 1 inch thick steel plates could be warped .05 inches apart before a preload torque is applied. Therefore, the entire theoretical preload (as calculated by the turn of the nut inserted) when applied, would do nothing more than draw these plates together. Also, dirt, rust or other compressible impurities between the plates could cause troubles.

The angular perpendicularity of a hole drilled into the plates could also be off. The washers used may not be flat (and it would not take much rotation to throw them off .010 inches). Since there are so many variables in bolt torquing, it would be dangerous to suppose that this "turn-of-the-nut" method could be an accurate one.*

Q. 31 When should the "turn-of-the-nut" method be used?

A. 31 Around heavy steel construction only, and where the bolts are assembled only once. The bolts must also have a high percentage elongation to provide some forgiveness. If such bolts are over-torqued, they would not break (neither the bolt nor the nut) because the steel used would generally be a low carbon steel.

Only in a case where the construction steel foreman knows the length of the bolts, the condition of the materials, etc., and where he has used a good calibrated torque wrench to check out his approximate turn of the nut calculations, should this method be used. Although this method is better than no method at all, unfortunately with the advent of higher strength alloy bolts in the steel construction industry, it is questionable whether the accuracy is now good enough for the job required. In light of the recent building failures which have been traced to bolted joint problems, every torque-up job should be evaluated based on its particular parameters.

There are some specific references for using this "turn of the nut" method, all of which should be reviewed before using the method. These include the following:

Helpful Hints, Russell, Burdsall and Ward Bolt and Nut Company, 1971, p. 15.[19]

Fastener Facts Bulletin, No. 35A, Bethlehem Steel Corporation, Industrial Fastener Sales, March, 1967.[20]

The Turn-of-the-Nut Method, M. D. Hoza, "Fastening and Joining", Assembly Engineering, January, 1967.[21]

*Punched washers are not flat and they sometimes flatten out or dig in when torqued. Lock washers are particularly bad.

Reader Feedback: The Turn-of-the-Nut Method, Jack Wilheld, "Fastening and Joining", Assembly Engineering, April 1967.[22]

Turn-of-the-Nut Method... Seven Simple Steps, E. F. Ball, Assembly Engineering, August, 1967.[23]

Reader Feedback: The Turn-of-the-Nut Method, Roger Hansen, "Fastening and Joining", Assembly Engineering, April, 1967.[24]

Bolt Preload--How Can You be Sure It's Right?, A. S. Cornford, Machine Design, March, 1975.[25]

Q.32 List other methods for limiting or controlling the preloads in a torqued bolt.

A.32
- Twist off nuts which fail when bolts are over-torqued.
- Slip heads and nut collars which slip when required torque is reached.
- Bolt heating which applies a predetermined preload when installed in a structure.
- Hydraulic tensioning which controls the load until the nuts are run up.
- Monitoring strain gages mounted on the bolt (usually within the head).
- Micrometer measurement of bolt lengths to determine elongations.
- Built-in strain measurement.
- Load indicating washers used to indicate total load.

Most of these are special devices (see Table 2-1).

[redacted] 37

	Requires special components	Requires special tools	Low cost	Sensitive to misalignment	Reusable	Needs loosening to check	Special access requirement	Minimum size limitation	Single load value	Requires stress calculation	Can be used in confined space	Requires extra bolt length	Can give remote indication	Sensitive to variation in thread length
Torque Measurement		x	x		x					x				
Twist-off Nut	x		x			x			x		x			
Turn-of-Nut			x	x	x	x				x	x			x
Bolt Heating	x	x		x	x	x		x		x				x
Hydraulic Tensioning		x			x		x	x				x		
Strain Gage	x			x	x			x		x	x		x	
Micrometer Measurement	x		x		x		x			x				x
Built-in Strain Indicator	x				x			x		x	x		x	
Load Indicating Washers	x			x	x				x		x	x		

Selecting the Right Tension Control Method

Table 2-1 (From Reference 7)

38 [redacted]

[redacted]

SECTION III

The Effects of Various Bolt Tightening Methods on Bolted Structure Stresses

Q.33 Now that a method has been established for calculating the load/torque relation for bolt-nut combinations, what is the next most important consideration?

A.33 It is necessary to understand exactly what happens to the bolt and to the structure as the nut or bolt is tightened. The clamped assembly deflections and stresses must be thoroughly understood.

Q.34 From figure 14, what is the step-by-step development of an understanding of the bolt tightening process?

A.34 To understand the bolt tightening process, separate the bolt and nut from the assembly as shown in figure 14a; next, stretch the bolt itself to its final position (14b); compress the frame together to its final position (14c); then reassemble the two parts together (14d) in their final positions as if the nut had been used to tighten up the combination. Next assume an external pull load is applied to the frame and bolt assembly (14e) and observe the variations in deflections and stresses in the parts.

In step (14a) the frame is shown split apart and the nut on the bolt is turned up to its final position (i.e., to a point at which it would end up if the nut had been turned up on the frame). Two horizontal lines marked "A" are shown on figure 14 which locate the starting point of bolt extension for reference.

Next visually stretch the bolt to a position where it would end up if it had been tightened up on the frame. The value Δ_1 represents the bolt elongation at each of its ends.

In (14c) compress the frame to a length $2\Delta_2$ (the amount which it would have deflected if the bolt had compressed the frame). Notice from the two horizontal beginning lines marked C that the frame has compressed to the bolt point or to the lines marked B. Further recall that the bolt had previously been elongated to these lines B, and therefore both the frame and bolt have ended up at the same horizontal positions.

While visually holding the bolt elongated and the frame compressed, simply envision that the bolt is placed back into the frame; assume now the bolt is holding the frame in compression, and that the frame (in compression) is keeping the bolt in tension (14d). Later it will be shown that in general, Δ_1 will not be equal to Δ_2.

The relative stiffness of the bolt to that of the frame will determine the actual differences between these two deflections.

Now if an external load is applied to the frame (14e), and it tends to pull the frame apart, the frame will extend an amount (Δ_3) at each of its ends. In order for this frame to elongate, the bolt must also stretch along with it. Thus this additional load applied to the frame will cause an additional extension strain in the bolt region and a tensile load increase in the bolt. Notice further as the frame stretches, that the frame parts between the bolt grip are not picking up more load, but losing it.

Finally, if the external load on the frame is increased to a large enough value, then the load in the central part of the frame will be reduced to zero. A gap will appear between the two sections of the frame and the bolt will be supporting all of the tensile load across the gap section.

Q.35 Determine the deflection of the bolt and the frame after an external load is added to the frame.

A.35 It is known that the tensile force in the bolt must be equal to the compression force in the frame, since there are only two members opposing each other. It is not known how much total deflection is in the bolt and how much is in the frame. This can be determined by a brief review of strength of materials to evaluate what happens to a member in tension or compression.

40 N

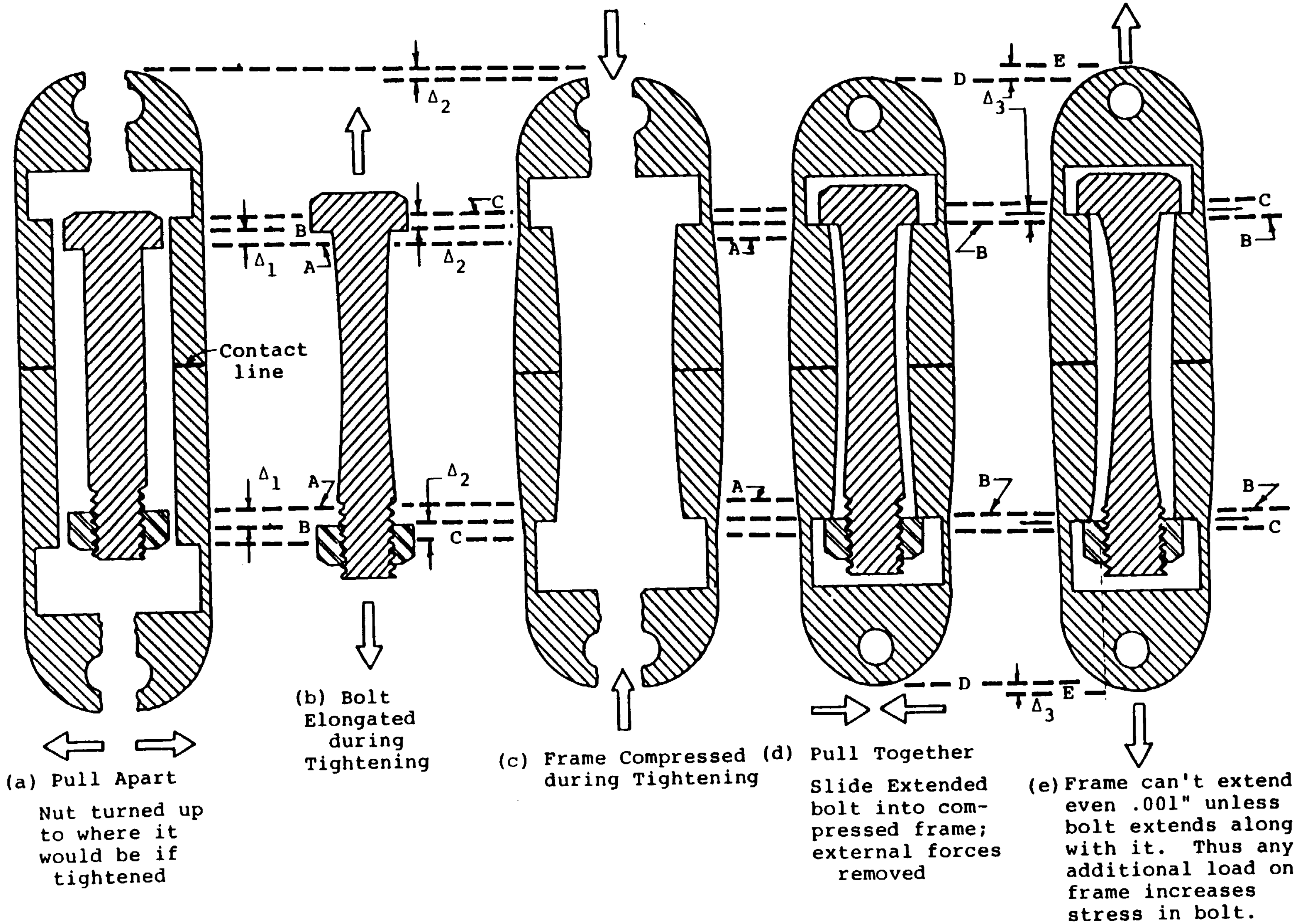


Figure 14 - Development of Tightening Method

$$\text{Let } K = \text{stiffness} = \frac{\text{Force}}{\text{elongation}} = \frac{F}{\Delta L} \qquad (4)$$

We can calculate the stiffness K, by the definition of Young's Modulus:

$$E = \frac{\text{stress}}{\text{strain}} = \frac{\text{Force/Area}}{\text{Elongation/Length}} = \frac{F/A}{\Delta L/L} = \frac{L.F}{\Delta L.A}$$

we find:

$$\Delta L = \frac{\text{Length x Force}}{\text{Young's modulus x Area}} = \frac{L.F}{A.E} \qquad (5)$$

For example, the bolt stiffness K_b is,

$$K_b = \frac{F}{\Delta L} = \frac{F}{L.F/E.A} = \frac{EA}{L} \qquad (6)$$

As an example, figure 15 shows an element of Area A and Length L under a tensile force F, and figure 16 shows the corresponding force-deflection diagram.

The stiffness of the bolt K_b will be the slope of this load deflection diagram shown in figure 16.

The stiffness of the frame K_f is the slope of the force deflection diagram for the frame in compression.

It can be observed from equations (5) and (6) above that if Young's modulus "E" is increased by changing materials (say from aluminum to steel), then the stiffness of the part also increases and the corresponding deflections go down.

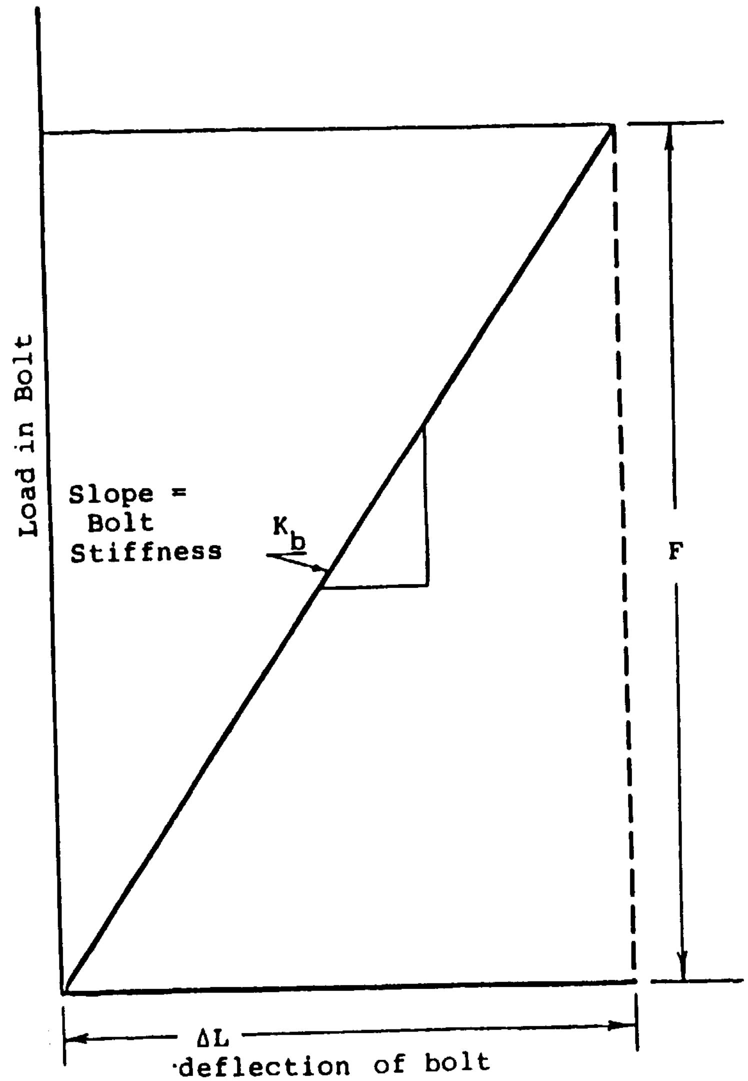
$$E = \frac{\text{Unit Stress}}{\text{Unit Strain}}$$

$$\text{Let } K = \text{Stiffness} = \frac{\text{Force}}{\Delta L} \qquad (4)$$

$$\Delta L = \frac{\text{Length x Force}}{\text{Young's Modulus x Area}} = \frac{LF}{EA} \qquad (5)$$

For example, the bolt stiffness is

$$K_b = \frac{F}{LF/EA} = \frac{EA}{L} \qquad (6)$$



Force deflection diagram for element of Figure 15.

Figure 16

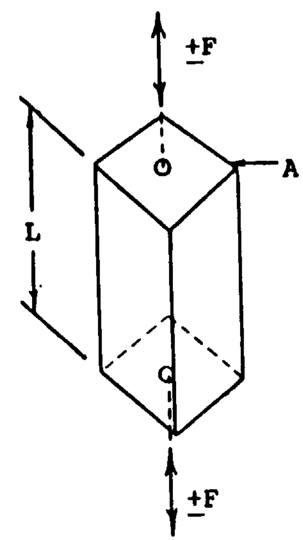


Figure 15 - Element which elongates under a tensile force

43 [redacted]

As the area of a part A is increased, the stiffness K also increases. As the length of part L is increased, the stiffness goes down. Applying this information to a bolt and nut assembly, we find the area of the frame is usually larger than the area of the bolt. Thus the frame is usually stiffer than the bolt for the same materials.

By making the Modulus of Elasticity (E) of the parts lower (i.e., changing from steel to aluminum), the joint will be less stiff. It will deflect more and consequently absorb energy better.

If the bolt were steel and the frame aluminum, the bolt would be three times as stiff as the frame, provided that they both had the same effective areas.

In most cases the grip length of the bolt is the same as the engagement length of the frame. However, in special cases, methods have been developed with springs, washers, etc. which vary the effective length.

As in figures 17a and 17b, the frame is generally stiffer than the bolt (because its load deflection slope is steeper). Now refer again to figure 14. The bolt in (14b) had to be pulled further apart than the compression deflection of the frame (14c). Further, if they were not pulled apart in proportion to their stiffnesses, the bolt and frame would adjust themselves to a proportional deflection position after the bolt load was released onto the frame. Figure 18 illustrates this fact.

It can be seen that there are a great many variables employed in the analysis of bolt and nut construction. These variables have such large exponents (such as 30,000,000 psi for the modulus of steel) that it becomes impossible to read a lengthy report development of the bolt problem and maintain an understanding of these variables as well. Thus, some parts of this report are written, not to explain the precise magnitude of these variables, rather to show the balance of all of these variables with sketches so that the design engineer can mentally visualize and calculate the entire problem at one time. To do this, it has been necessary to use fictitious values of Young's modulus "E", deflection, etc.

In this calculation, hypothetical numbers are used only to illustrate the principles. From figure 19, the stiffness of the bolt K_b is assumed to be 1000 lbs/in. The bolt will then deflect 1 inch for every 1000 lbs applied. The total load on the bolt is assumed to be 4000 lbs. Thus the deflection of the bolt corresponds to a 4 inch extension relative to its free length.

Similarly the stiffness of the frame is assumed to be 2000 lbs/in. Since the total force on the bolt must equal the total force on the frame as they react against each other, the force on the frame would also have to be 4000 lbs. If the frame deflects 1 inch for each 2000 lbs applied, it would move 2 inches (relative to its free length) under a total load of 4000 lbs.

Note however that deflections are generally much smaller than this in bolts, and the deflections in this example have been exaggerated to see what proportion each part takes.

Now that a clear picture of the relative deflections of both the frame and the bolt is available for equal preloads, an exterior load can be applied to the frame. The problem then is to see how far the bolt deflects and how far the frame deflects under this added load of 2000 lbs.

This external load is now applied to the pre-stressed bolt and frame joint. Remember that this load is applied initially to a frame which is already under a compressive load and has deflected.

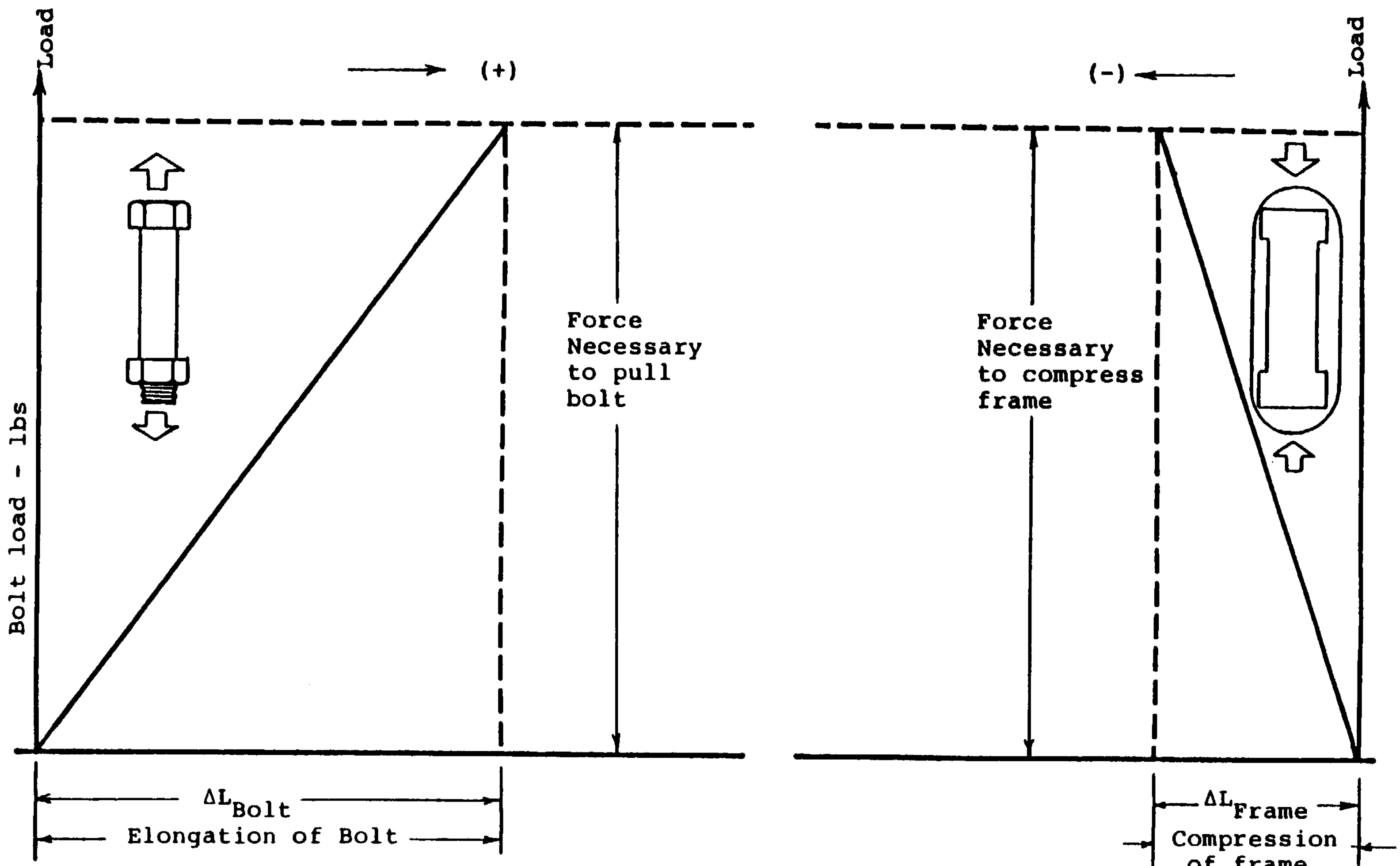


Figure 17a
Force Deflection Diagram

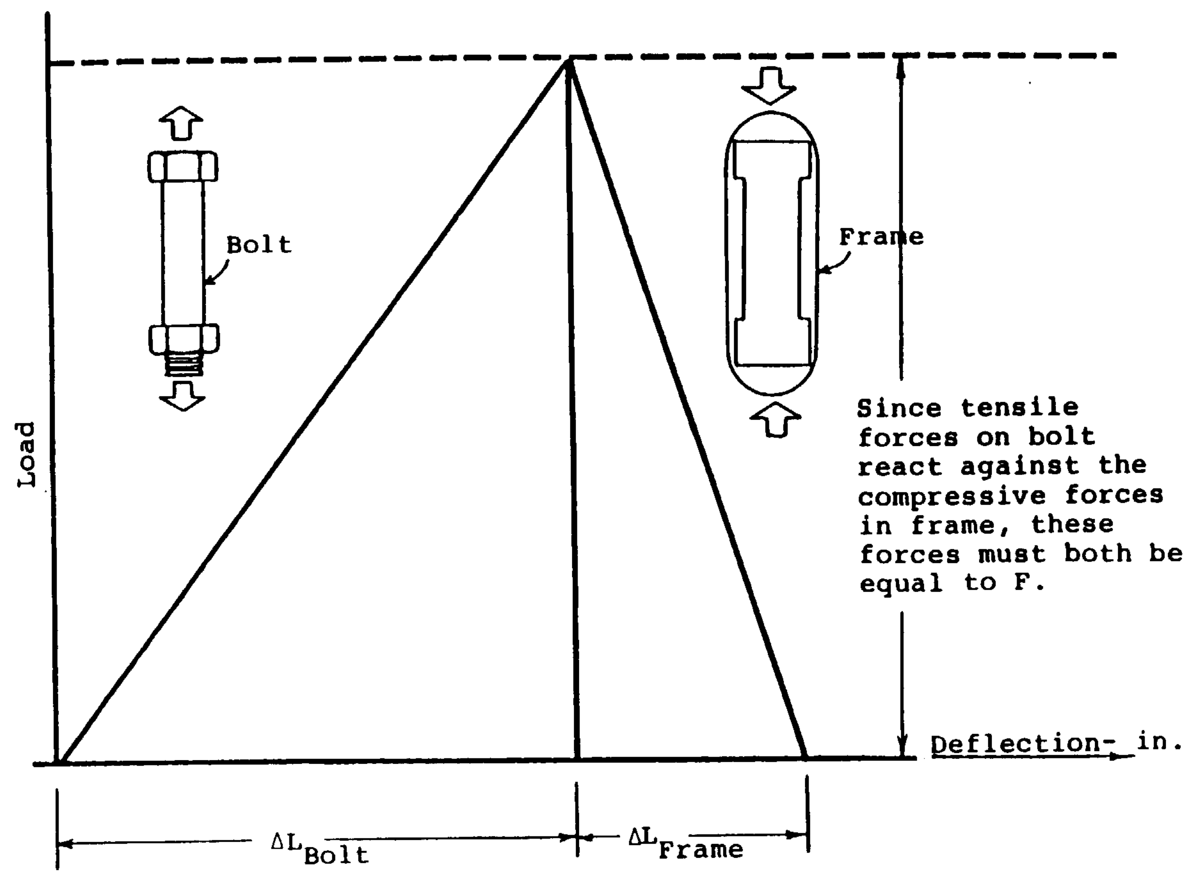


Figure 17b

Combined Force Deflection Diagrams for Frame and Bolt

47 [redacted]

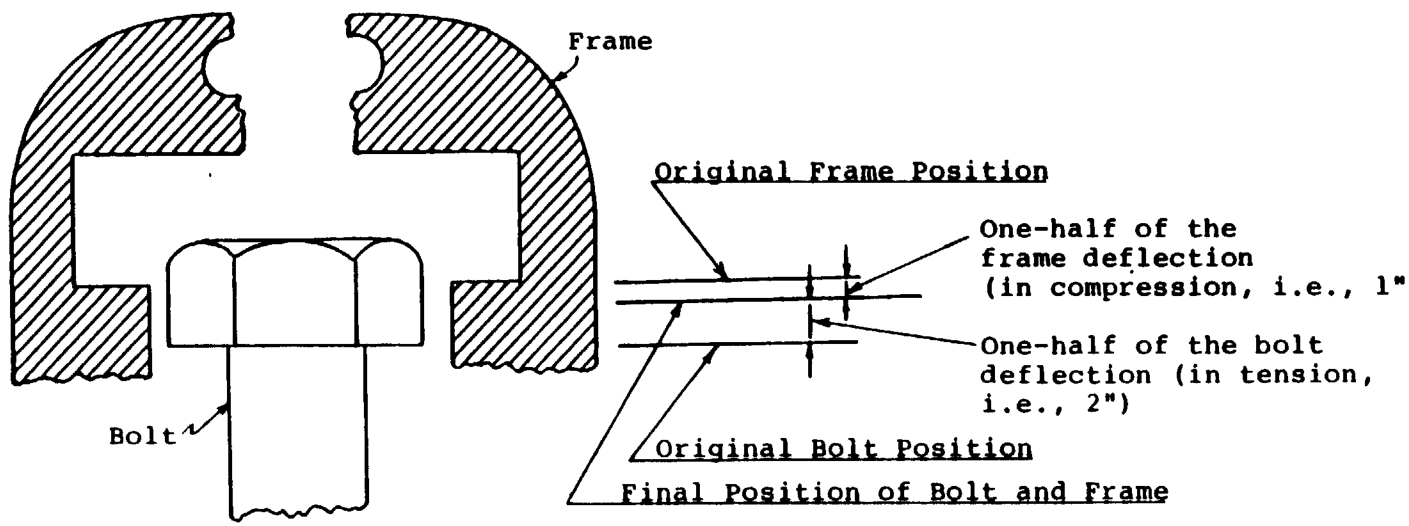
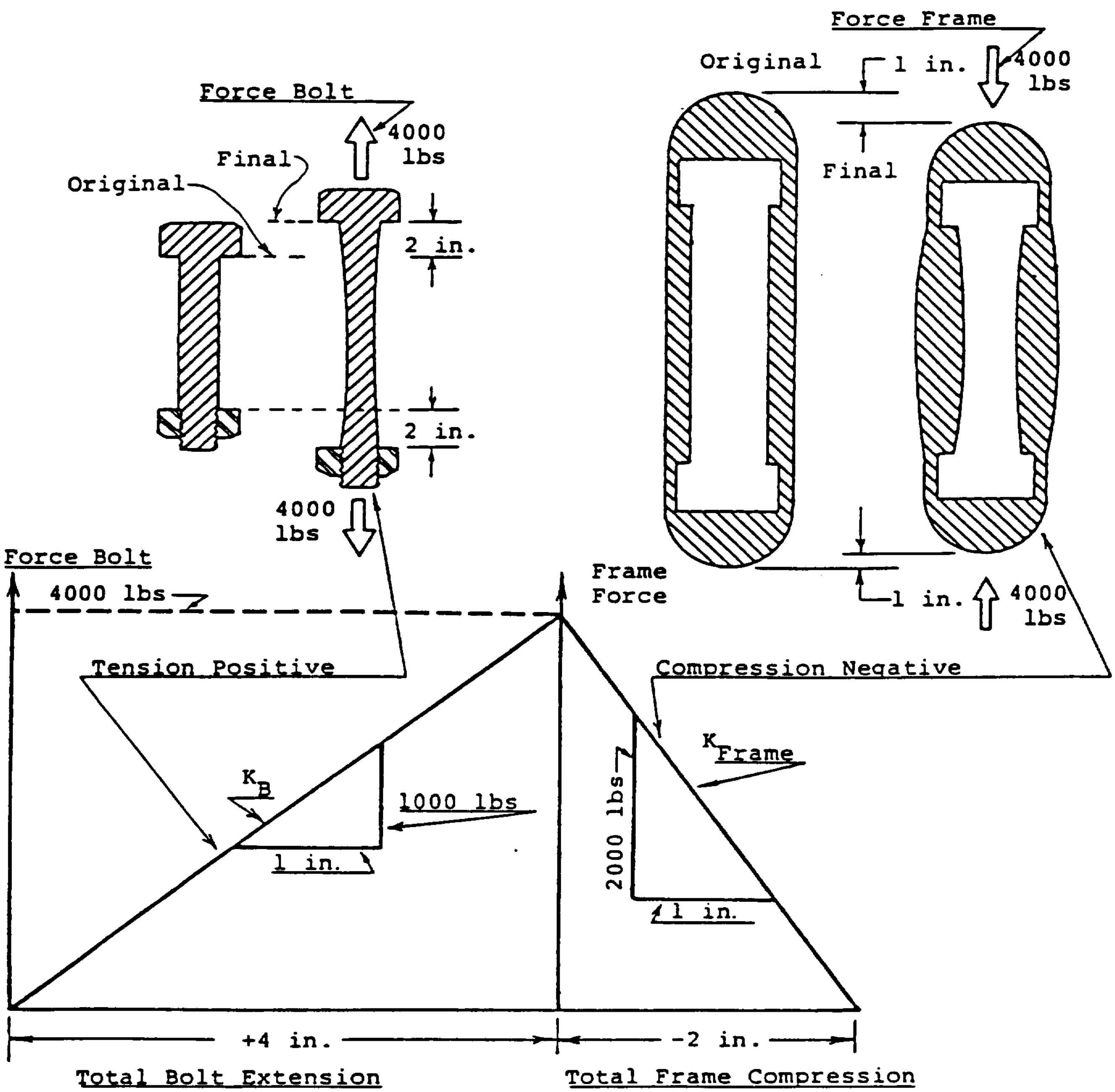


Figure 18 - Frame and Bolt Positions



Note:
- o The bolt extends 1000 lbs/in or 4 in (relative to its free length when a 4000 lb load is applied.
- o Frame compresses 2000 lbs/in or 2 in (relative to its free length) when a force of 4000 lbs is applied.
- o Under equilibrium the bolt in tension compresses the frame and the frame in compression extends the bolt.
- o The frame is twice as stiff as the bolt and thus it moves one half as far. (Bolt moves 4 in and frame moves 2 in).

Figure 19 - Bolt Extension and Frame Compression Example

As the external load of 2000 lbs is applied, it is not readily known how much of the load the bolt will take. However, the rate of load in the bolt must still be 1000 lbs/in. Thus, if the frame were not there, the bolt would move 2 inches under this load.

All of the externally applied loads from the top point of the frame (point A in figure 20) are supported by the frame itself between points A and B. Between points B and C, part of this load is supported by the bolt and part by the frame.

Recall that for this case the frame and the bolt are both under 4000 lbs preload, and that the bolt has elongated 4 inches (relative to its free length), while the frame has compressed only 2 inches. (Figure 21.)

As the external load is applied, the frame which was originally compressed loses some of its compression. The rate at which the frame loses this load is the same rate at which it was compressed. In other words, the frame always has the same physical stiffness. The only thing not known is how much of the external load the frame will take.

The rate at which the bolt will take up its load and the rate at which the frame will give up some of its load are known. Only one other thing is known -- the amount of deflection given up by the frame will be taken up by the bolt. Before and after the additional load of 2000 lbs is applied, the bolt head and nut are in constant contact with the frame. The above observations are shown quantitatively on the load-deflection curve of figure 22.

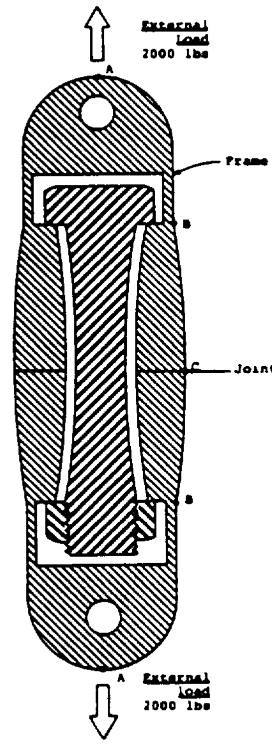


Figure 20 - External Loads

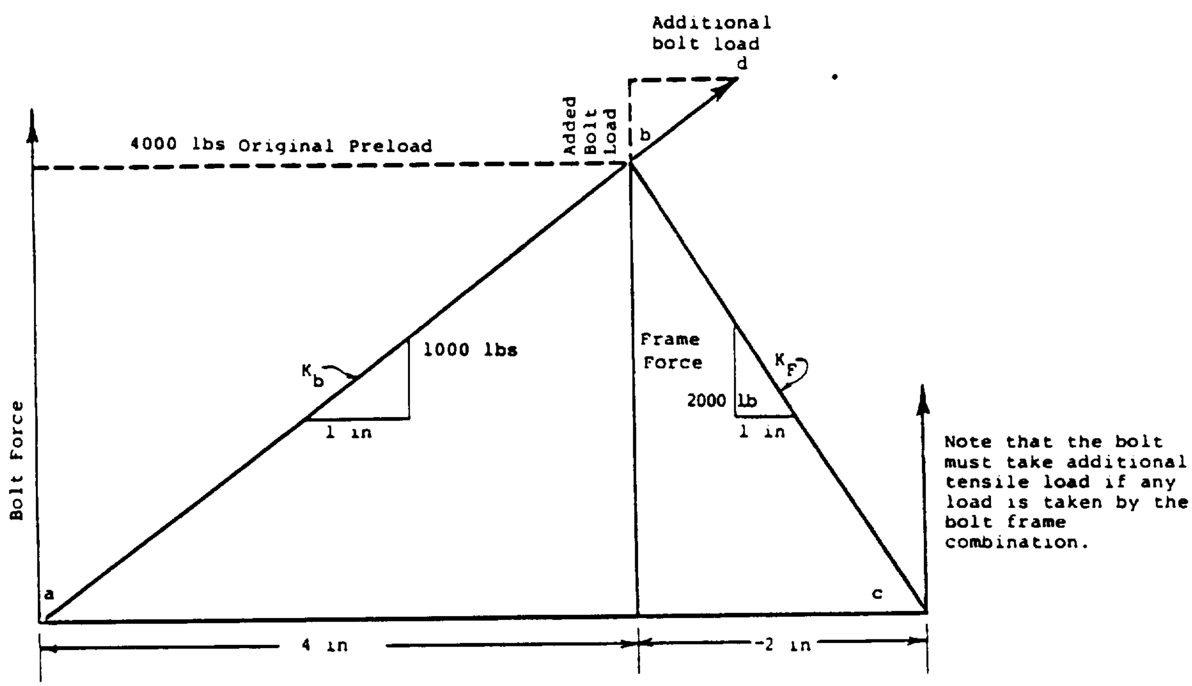


Figure 21 - Load/deflection Diagram for Pre-stressed Bolt and Frame

Figures 22 through 24 show the proportions of the original compressive load lost in the frame (f-d) and that gained by the bolt (c-f), along with the deflections picked up by the bolt and lost by the frame (b-f). An expression is developed in Appendix A-4 which provides a convenient expression of the additional bolt load "$Load_b$" and frame load "$Load_f$" so that

$$Load_b = \text{bolt added load} = F_e \frac{1}{1+1/R}$$

$$Load_f = \text{frame load added} = - F_e \frac{1}{1+R} \tag{7}$$

where F_e = total added load, and $R = \frac{\text{stiffness of bolt}}{\text{stiffness of frame}} = \frac{K_s}{K_f}$

Given the spring constant of a bolt as K_b = 1000 lbs/in, and given the spring constant of the frame as K_f = 2000 lbs/in., and the additional external load on frame is force F_e (2000 lbs), the ratio $R = K_b/K_f = .5$, and from equation (5), the bolt load addition is

$$Load_b = \frac{1}{1+1/R} F_e = \frac{2000}{3} \text{ lbs} = 667 \text{ lbs}$$

The frame internal load can be found by the formula:

$$Load_f = - \frac{1}{1+R} F_e = - \frac{2000 \text{ lbs}}{1.5} = - 1333 \text{ lbs}$$

Since the bolt and frame are originally pre-stressed to 4000 lbs each, the final bolt load is 4000 lbs + 667 lbs = 4667 lbs. The frame final load is 4000 lbs - 1333 lbs = 2667 lbs. (See Figure 24).

The change in the deflection of the frame after the external load F_e is added (beyond the original deflection) is:

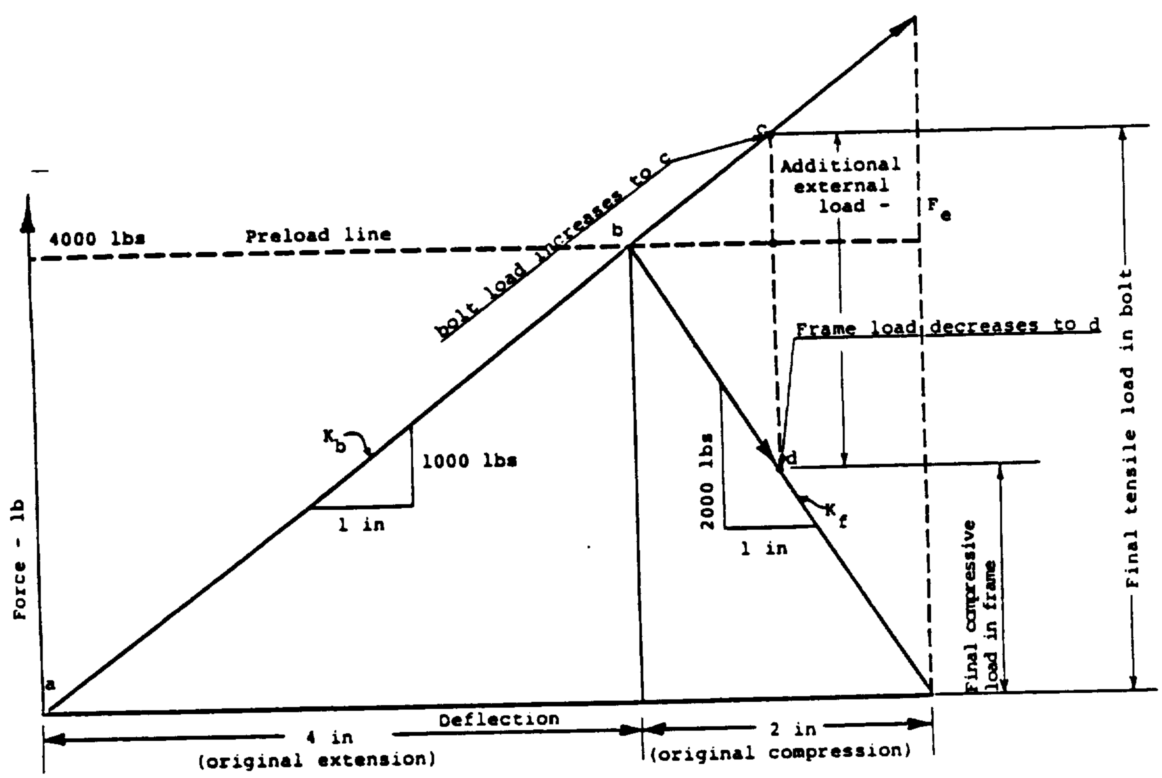


Figure 22 - Load/deflection Diagram for Pre-stressed System With Added Tensile Load to Assembly

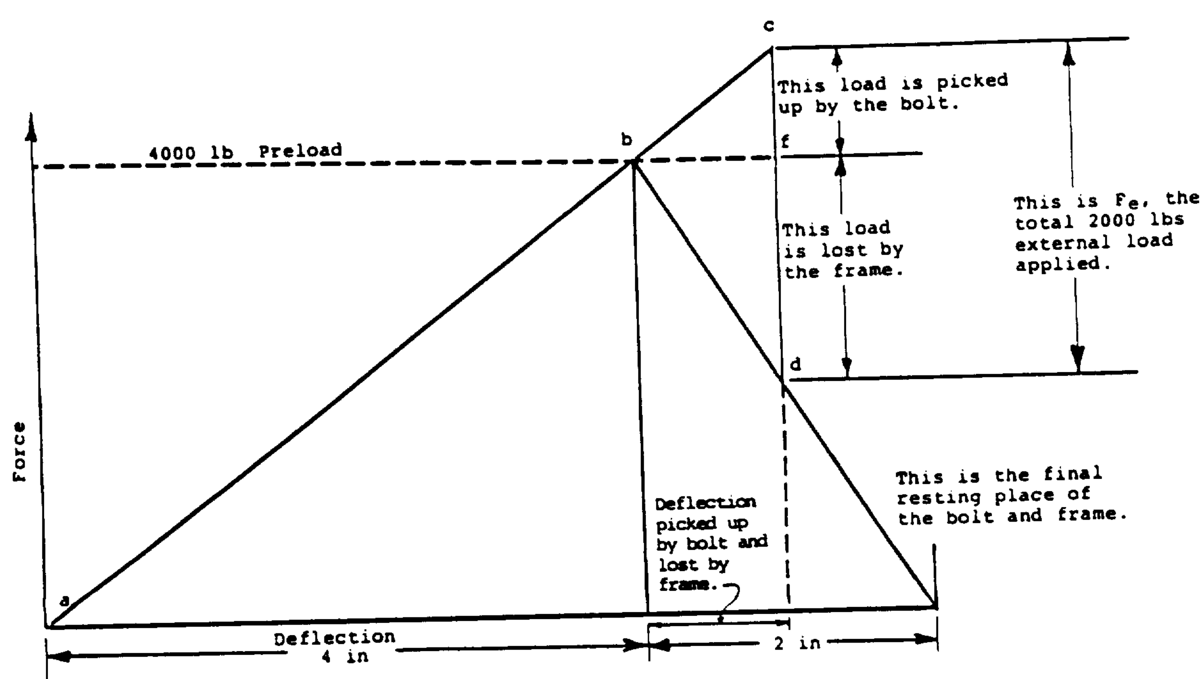


Figure 23 - Deflection Diagram for Pre-stressed System With Added Tensile Load

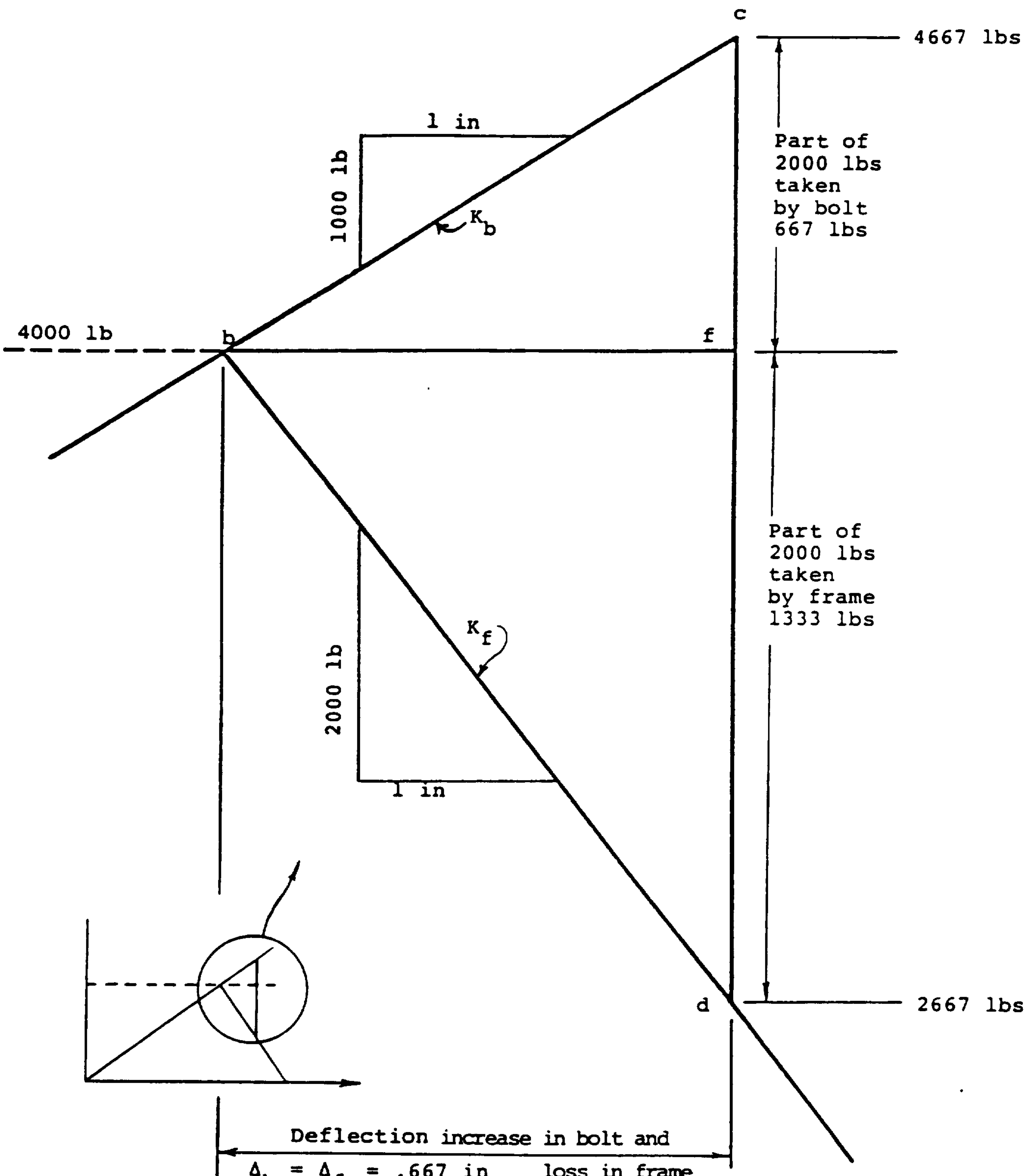


Figure 24 - Load Distribution Diagram

54N

$$K_f = \frac{Load_f}{\Delta_f}$$

$$\Delta_f = Loads_f/K_f = -\frac{1333}{2000} = -.667 \text{ inches} \qquad (8)$$

Since both the bolt and frame are in contact, Δ_b is the same. The load in the bolt has therefore increased to 4667 lbs, the frame load is 2667 lbs.

The deflection of the bolt can be calculated:

$$Load_b = K_b \Delta_b$$

$$\Delta_b = Load_b/K_b = 667 \text{ lbs}/1000 \text{ lbs-in}$$

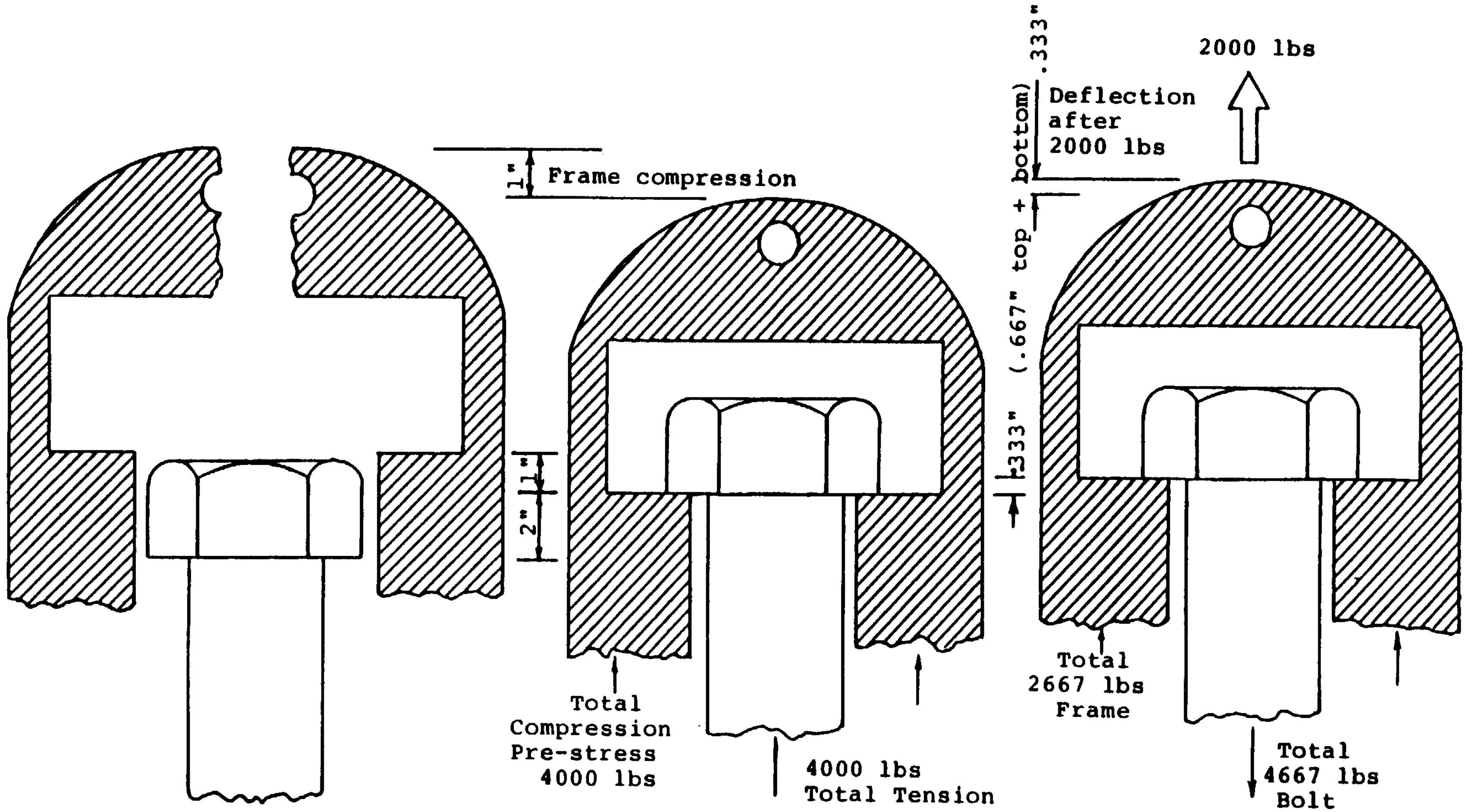
$$\Delta_b = .667 \text{ inches}$$ (see Figures 24 and 25)

Q.36 Now the loads and deflections on both the bolt and the frame can be calculated while preloading the bolt. Further, the additional loads and deflections on the bolt and the frame when additional tensile loads are added to the frame can be determined. How does determining these loads relate to the bolt's internal stresses?

A.36 Loads are always related to stresses as some of our original questions pointed out. The stress distribution will vary from thread to thread and under the bolt head, as shown in figures 2a, 2b and 3 (Section I). Even though the total load is known, this does not mean that the locations or exact values of critical stresses are known.

Q.37 Then why was so much time spent studying the bolt loads, frame loads and deflections necessary?

A.37 For design purposes, it is necessary to compare different types of bolt-frame conditions, to select the best design. While the exact value and distribution of bolt stresses are not known, the comparative values of these stresses are a function of the average stress, which is known.



Initial Position of Bolt and Frame

Position of Bolt and Frame after Bolt is Torqued Up.

Position of bolt and frame after 2000 lbs additional are added to frame.

Figure 25 - True Deflection of Pre-Stressed Bolt with Additional Load

[redacted] 56

Therefore it is possible, for example, to investigate the use of washers, gaskets, springs, etc. in a design to see if they increase or decrease the loads, and consequently the average bolt stresses. Although the exact stress values are not known, the best course of action to take in each of these designs can be selected.

will have different requirements. For example, a brittle bolt may need to move under shock conditions. If this is not permitted by the design, the bolt may suffer a brittle fracture at relatively low loads. If a ductile bolt moves too far, on the other hand, it may start to yield at its points of high stress during preload. Additional loads on this bolt would produce a permanent elongation and would not allow the bolt to return to its original position. This condition may even cause the nut to back off and permit the frame to "gap".

Q. 38 How do the use of washers in a design affect the bolt nut loads and their deflections?

A. 38 A typical bolt will be used to draw two plates together. Observe from figure 26 how the bolt compresses the plates and how the plate compressive force tends to extend the bolt. To quote from three excellent references, it is possible to determine what an effective frame area A_e might be:

"Untersuchungen Uber Die Sicherungseigenschaeften Von Schrauben Bindungen Bei Dynamisher Belastung"[26]

"Bolted Joints - How Much Give?", Robert E. Little, Machine Design, Nov. 1967[27]

"Simple Diagrams Aid in Analyzing Forces in Bolted Joints", Gerhard Meyer, Assembly Engineering, Jan. 1972[28]

In figure 27, the shaded areas in each cross section propose effective areas of compressive stress obtained from each of the above proposed reference methods. The shape shown in figure 27c is considered to be the most widely accepted.

Figure 28 shows the stress distribution over the effective area obtained by the axisymmetric finite element method.

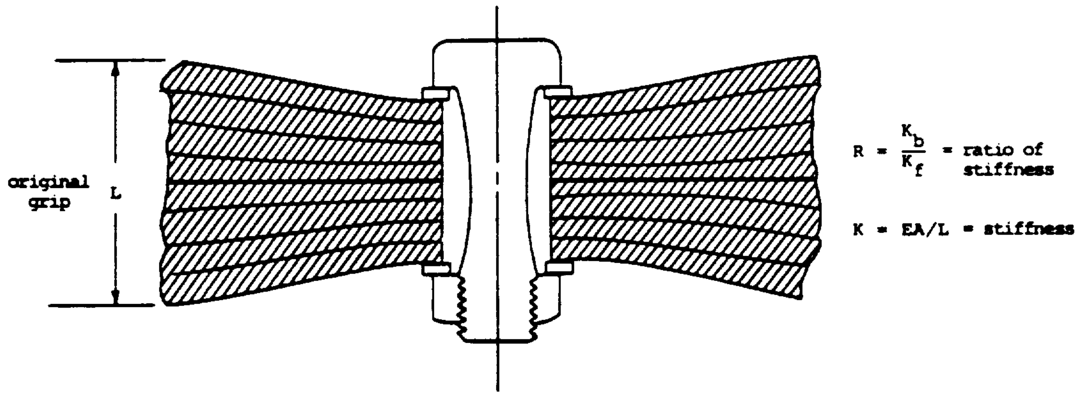
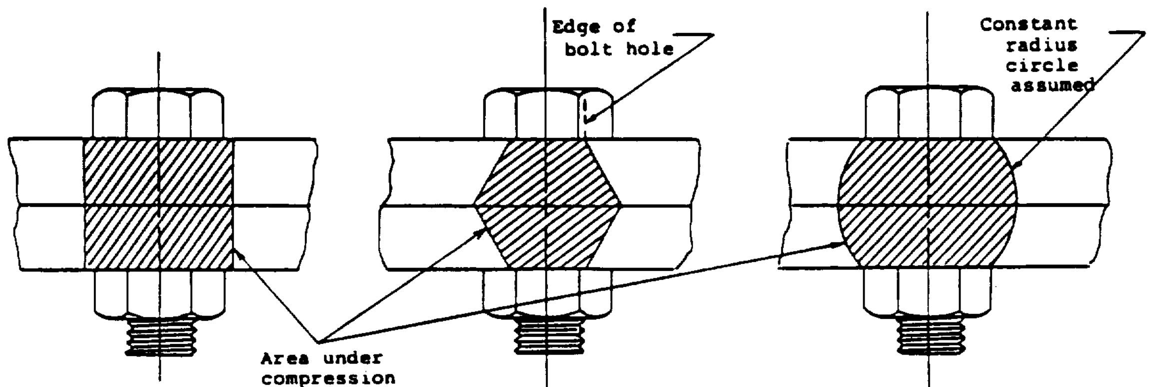


Figure 26 - Compressive Force in Plates Bolted Together



(A) (B) (C)

Figure 27a - Proposed Theoretical Areas A_e of Plate Under Stress

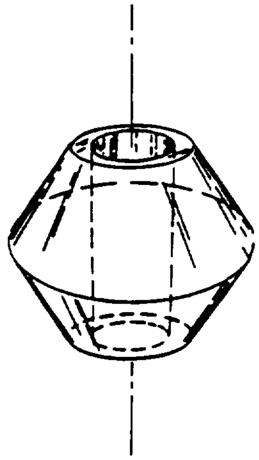


Figure 27b

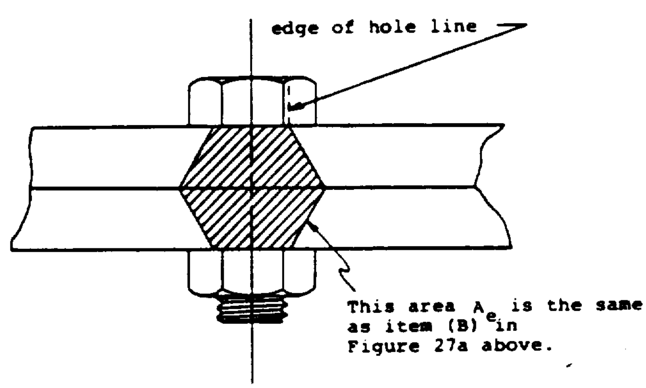
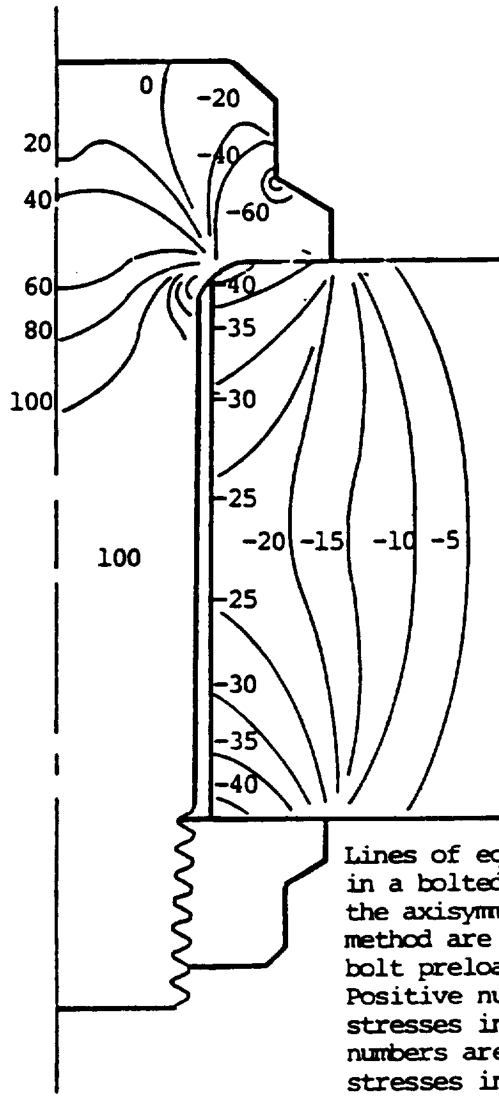


Figure 27c

Proposed Theoretical Areas of Plate
Under Stress

[redacted] 5B



Lines of equal axial stresses in a bolted joint obtained by the axisymmetric finite element method are shown for a 9/16-18 bolt preloaded to 100 ksi. Positive numbers are tensile stresses in ksi; negative numbers are compressive stresses in ksi.

Figure 28 - Stress Distribution in Bolt and Nut

While there is not an exact value for the frame area under compression, an approximate area can be determined that will be sufficient for study (see figure 29a and b). It can be seen how the area in the frame under compression is changed proportionally by adding a washer. The effect of a washer will increase the contact area of this frame (in proportion to the square of the ratio of the diameter of the bolt hole to the washer outside diameter). This washer must be sufficiently thick and strong in flexure so that it does not bend. It must also be extremely flat so that it distributes its load evenly over the entire contact area.

Therefore, if a washer is used, it is possible to re-evaluate the previous load deflection example curves to see the washer's effect (see figures 30, 31 and 32). The following design observations are made from these figures:

- o As the frame or plates are stiffened by increasing the effective area A_e under compression, the amount of total bolt load (due to an additional external load) increase is less. In figure 32 this increased load reduces from 667 lbs to only 416 lbs.
- o A washer can substantially increase the stiffness of a frame (i.e., from 2000 lbs/in to 3800 lb/in for a typical washer as shown in figures 31 and 32).
- o The greater the stiffness of the frame, the less will be its deflection due to preload. Notice that the deflection reduces from .667 in to .416 in with the addition of a washer in figure 32.
- o As the frame is stiffened by using a washer, the frame loss of preload with the application of an increased load is more pronounced. For example, the loss of only 1333 lbs for a design without a washer is compared to a loss of 1584 lbs when a washer is added (see figure 32).

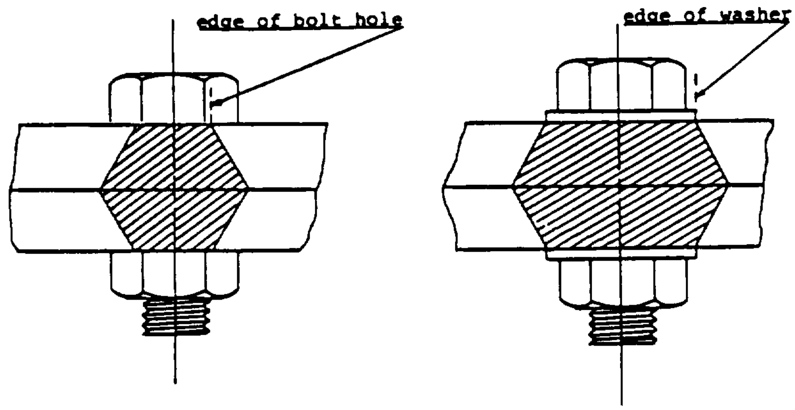


Figure 29 - Effect of Addition of Washer on Size of Plate Area Under Compression

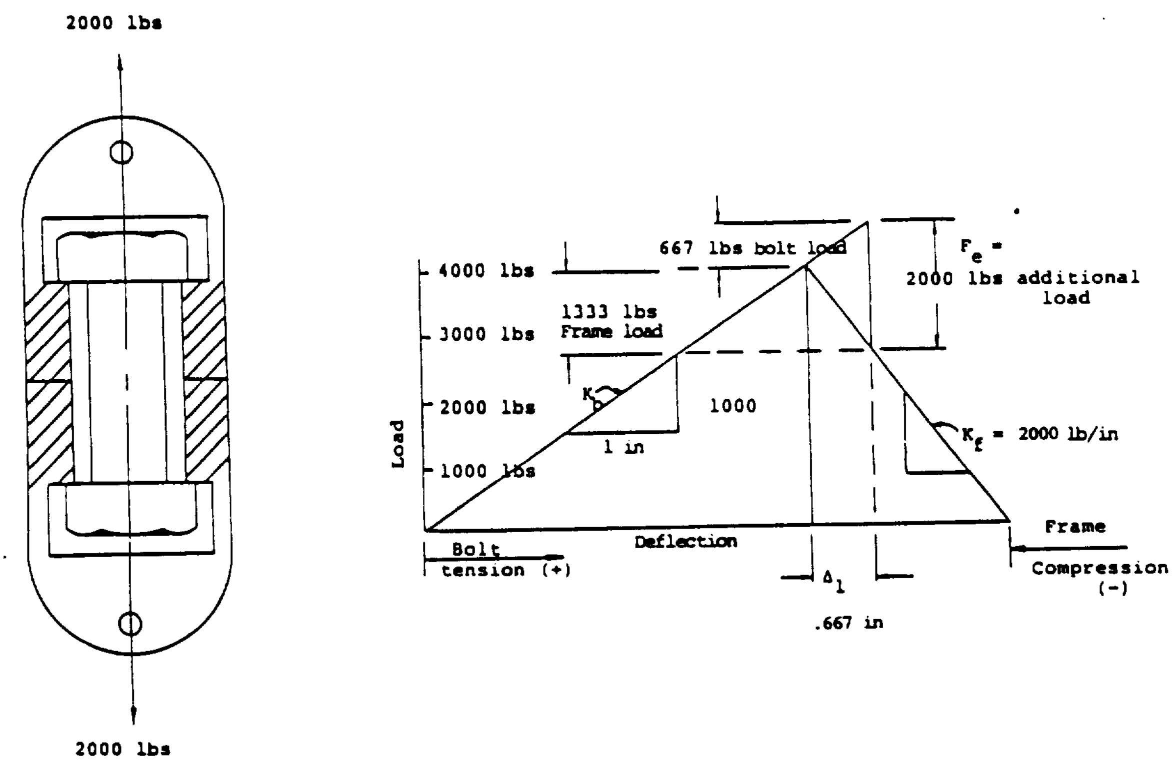


Figure 30 - Load Distribution in Bolted Frames
Under a 2000 lb Additional Load

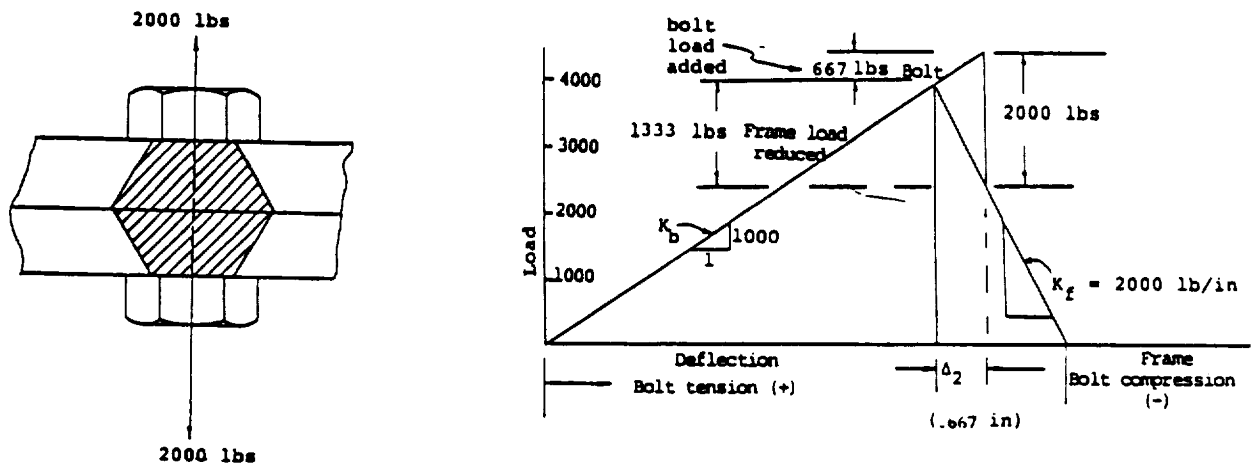


Figure 31 - Load Distribution in Two Plates Bolted Together Under 2000 lb Load

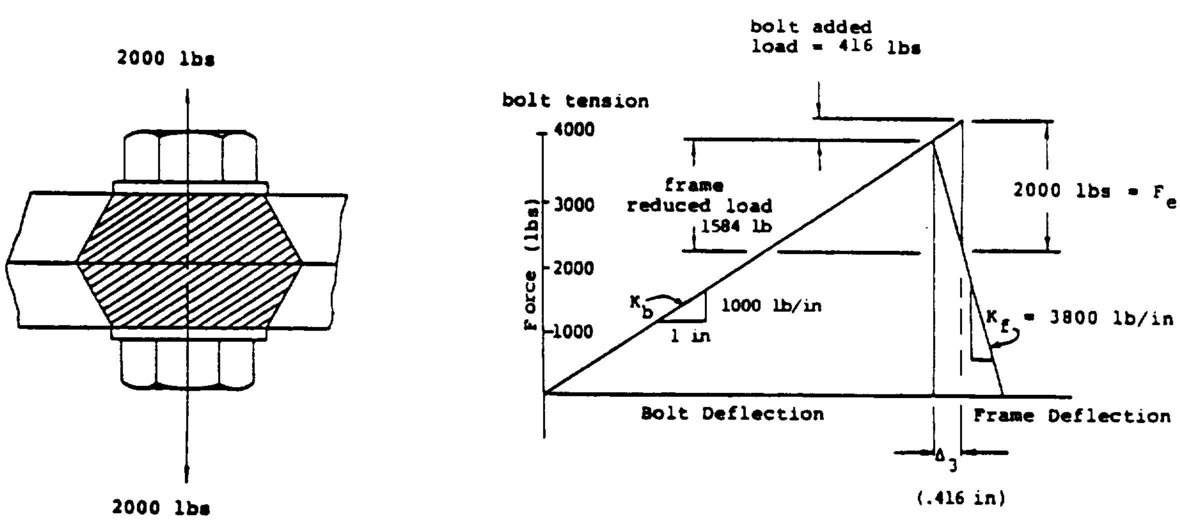


Figure 32 - Load Distribution Assuming a Washer is Added to Plates Under a 2000 lb Load.

62

- o An aluminum frame with a steel bolt would produce a small frame stiffness because of the relatively low Modulus of Elasticity of the aluminum frame. This frame stiffness would be improved with the use of a good washer. The same would be true with magnesium or plastic frames.
- o With high strength brittle bolts (i.e., bolts with low ductility) it is necessary to have some motion with any external load application. If a brittle bolt is short in its grip length, then its elongation will be low and brittle fracture will be more likely. Since the strain energy must be absorbed within that volume of bolt under strain, the shock-load capability must be increased for such a system by putting (for example) more than two washers under the nut to increase the bolt's grip. This design increases the length of the bolt grip and the effective frame length. This longer bolt grip can absorb more strain energy and therefore absorb a shock load better.
- o In some cases designers will call for "necking down" a bolt to reduce its stiffness.* This practice allows the bolt to elongate further under an applied load and therefore it can absorb more energy. From the diagrams of 30, 31 and 32, it can be seen that the bolt deflects further in figure 30 (due to the thin wall frame) and absorbs more energy than it would in figure 32. Combine this fact with the use of more washers to increase bolt length, and there is a double capacity for absorbing energy. Furthermore, a bolt may be made to deflect an additional amount by "necking down".

*Necking down requires machining down the diameter of the bolt shank between the head and the threaded portion of the bolt.

However, the use of washers increases the grip length of the bolt and it can undergo large deflections without yielding or failure.

Q.39 How are gasketed joints torqued up?

A.39 To examine the problem, first we must find the stiffnesses of both the bolt and the frame. The frame stiffness is a combination of both the gasket material and steel gasket retainer material.

The effective stiffness of the gasket/steel frame can be determined by a simple calculation. Assuming, for example, that the frame is steel and the gasket is a material with 1/10 the Young's modulus of the steel, the stiffness of each part could be calculated. Assume the following moduli and lengths for the configuration shown in figure 33:

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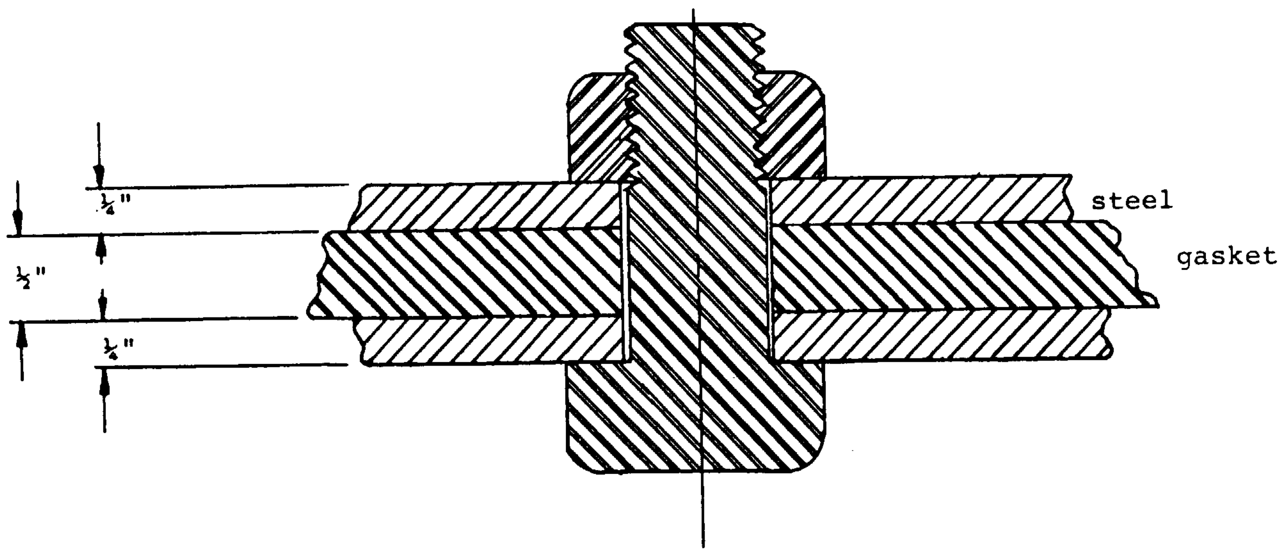


Figure 33 - Gasket Joint

(Assume, for convenience, that the modulus of steel is 30* and that of its gasket is 3*.)

$$E_{steel} = (E_s) = 30 \text{ psi}$$

$$Length_{steel} = (L_s) = 1/2 \text{ in}$$

$$E_{gasket} = (E_g) = 3 \text{ psi}$$

$$Length_{gasket} = (L_g) = 1/2 \text{ in}$$

Recall from equation (4) we defined "stiffness" K as:

$$K = \text{Stiffness} = \frac{\text{force}}{\text{Elongation}} = \frac{F}{\Delta L}$$

The compliance C is defined as the reciprocal of the stiffness, or $C = 1/K$.

$$C = \text{Compliance} = \frac{1}{K} = \frac{\Delta L}{F}$$

*Note: It is known that the modulus of steel is actually 30×10^6 psi and the one million factor has been deliberately ignored. Our resulting deflections should be divided by one million.

The total compliance of the steel and its gasket is:

$$C = 1/k = 1/k_s + 1/k_g$$

Next, substitute in actual numbers from figure 33, and assume the preload P on the assembly to be 10 lbs, then:

$$\Delta_s = \text{steel retainer deflection} = \Delta_1 = \frac{PL_s}{A_s E_s} = \frac{(10)(.5)}{(1)\ (30)} = .167 \text{ in}$$

$$\Delta_g = \text{gasket deflection} = \Delta_2 = \frac{PL_g}{A_g E_g} = \frac{(10)(.5)}{(1)(3)} = 1.667 \text{ in}$$

The total deflection of both parts would be:

$$\Delta_g + \Delta_s = 1.834 \text{ in} = \Delta_{frame} \quad \text{(This is for a 10 lb load.)}$$

Similar calculations can be found if a 1.0 lb preload were used. The reader should show that the steel deflection would be $\Delta_s = .0167$ and the gasket deflection would be $\Delta_g = .167$ in, or a total of $\Delta_{frame} = .184$. The final parts are compressed to the thicknesses shown in figure 35 for a 1.0 lb preload.

The stiffness of each part is:

$$K_s = \frac{A_s E_s}{L_s} = K_s = \frac{(1)(30)}{.5} = 60 \text{ lb/in}$$

$$K_g = \frac{A_g E_g}{L_g} = K_g = \frac{(1)(3)}{.5} = 6 \text{ lb/in}$$

Obviously the gasket is much softer than its steel retainer.

The effective stiffness of the combined parts is found from the reciprocal of the compliance:

$$C = \frac{1}{K_f} = \frac{1}{K_s} + \frac{1}{K_g} = \frac{1}{60} + \frac{1}{6} = \frac{1}{5.45} \ \frac{\text{in}}{\text{lb}}$$

$$K_f = \frac{1}{C} = 5.45 \text{ lb/in}$$

Therefore, the combined stiffness of the flange including its steel holder plates and gasket is K_f = 5.45 lb/in. Since this stiffness is known, a loading diagram similar to the previous examples can be drawn (see figure 34).

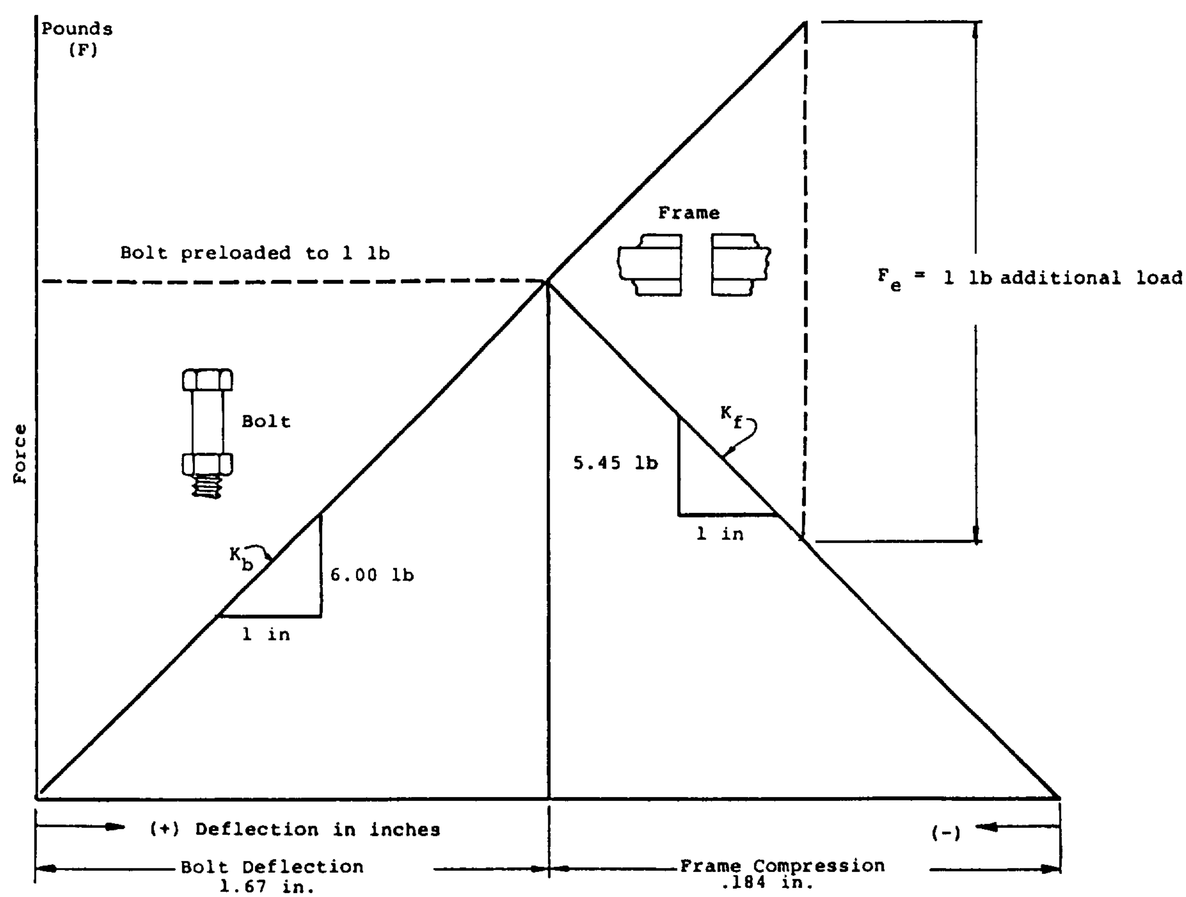


Figure 34 - Bolt and Frame Stiffness, Load and Deflection

The bolt stiffness was calculated as K_b = 6 lb/in, and the frame stiffness K_f = 5.45 has been found. The deflection of the bolt and the plate for a 1 lb load is found from:

$$\text{deflection } \Delta = \frac{\text{Force}}{\text{Stiffness}} = \frac{P}{K}$$

$$\Delta_{bolt} = \frac{1 \text{ lb}}{6 \text{ lb/in}} = .167 \text{ in}$$

$$\Delta_{frame} = \frac{1 \text{ lb}}{5.45 \text{ lb/in}} = .184 \text{ in}$$

See Figures 35a and b for reference in the following discussion. This deflection is measured from the original unloaded position of each part and occurs during preload.

Using the procedures illustrated previously, the frame and bolt loads can now be determined.

The ratio of bolt stiffness to frame stiffness (R) is:

$$R = \frac{K_b}{K_f} = \frac{6 \text{ lb/in}}{5.45 \text{ lb/in}} = 1.101$$

Using the expression from Appendix A4 for bolt load increase for an additional frame load F_e = 1 lb gives:

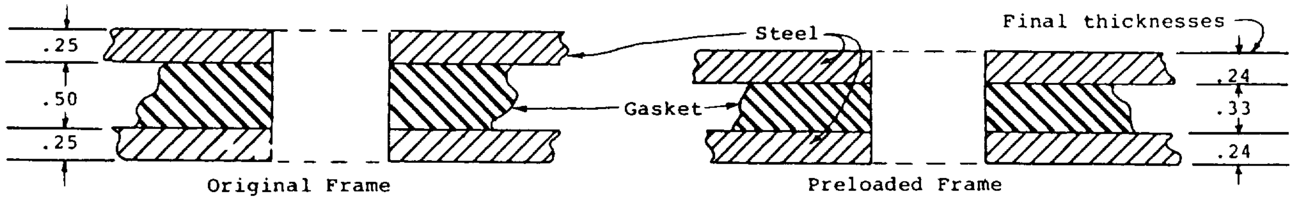
$$\text{Load}_b = \frac{F_e}{1+1/R} = \frac{1.0 \text{ lb}}{1+1/1.101} = .524 \text{ lbs}$$

$$\text{Load}_f = \frac{F_e}{1+R} = \frac{1.0 \text{ lb}}{1+1.101} = .476 \text{ lbs}$$

The deflection of the bolt or the frame can be found from equation (4).

$\Delta_b = F_b/K_b$, or for a bolt stiffness of K_b = 6 lb/in

$$\Delta_b = \frac{F_b}{6} = \frac{.524}{6} = .087 \text{ inches}$$



Final steel thickness .25 - .00835 = .24

Final gasket thickness = .5 - .167 = .333

Figure 35 - Frame Deflections

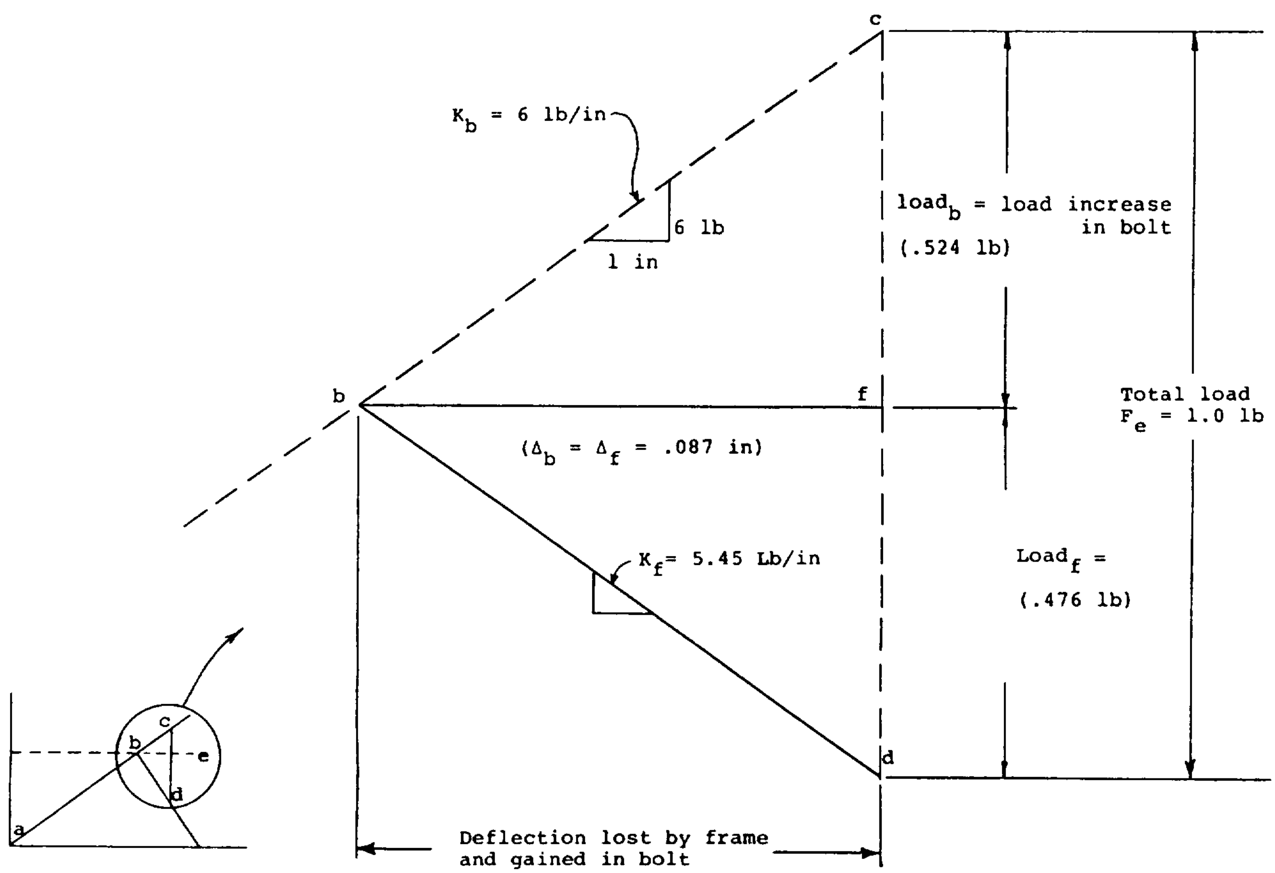


Figure 36 - Load Distribution Between Frame and Bolt

Similarly, for the frame stiffness found (i.e., $K_f = 5.45$ lb/in)

$$\Delta_f = \frac{F_f}{5.45} = \frac{.476}{5.45} = .087 \text{ inches}$$

Both the frame and bolt additional loads and deflections have been found. The separate deflections of the gasket and steel retainer parts of the frame must still be found. Therefore, the deflection of the gasket and steel plates after the extra load of 1 lb is applied are to be found. Recall the total added deflection of the combined steel and gasket was found to be .087 inches. It is also known that the total deflection was:

$$\Delta_s + \Delta_g = .087 \text{ inches}$$

The total force on the steel and gasket, (from equation 4 or figure 23) the stiffness/deflection relation were found:

$$K_f \, \Delta_f = F_f = .476 \text{ lbs}$$

Also, the values of the stiffness factors were found previously, i.e.

$$K_{st} = \frac{A_s E_s}{L_s} = 60 \; \frac{\text{lb}}{\text{in}}$$

$$K_{gasket} = \frac{A_g E_g}{L_g} = 6 \text{ lb/in}$$

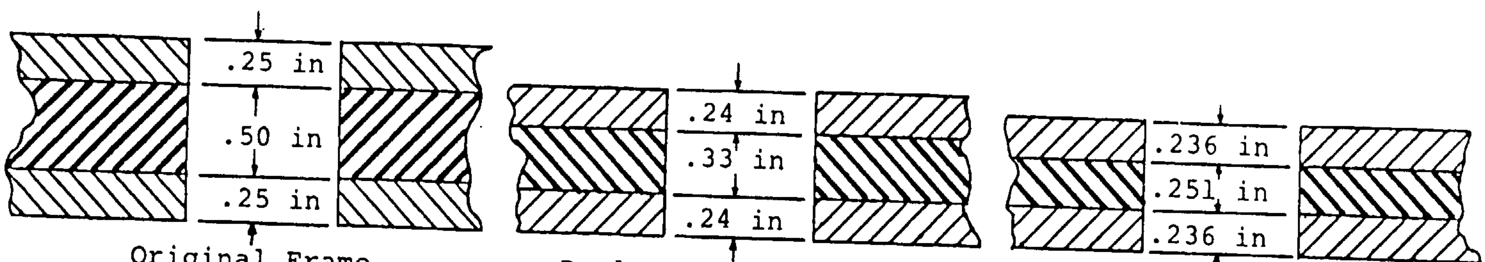


Figure 37 - Final Thickness of Gasket Joint

Recall from (4) that the stiffness K is related to a deflection by:

$$K = \frac{\text{Force}}{\Delta L}$$

$$\Delta L = \frac{\text{Force}}{K}$$

Since it is known that the forces on the steel and gasket materials must be equal (i.e., $F_s = F_g$), it is possible to find the deflection of each part:

$$\Delta_s = \frac{.476 \text{ lb}}{60 \text{ lb/in}} = .008 \text{ in} \quad \Delta_g = \frac{.476 \text{ lb}}{6 \text{ lb/in}} = .079 \text{ in}$$

The total deflection Δ = .008 + .079 = .087 inches. The final steel plate thickness would be .24 in - .004 in = .236 inches for each plate and the final gasket thickness would be .33 - .079 = .251 inches. Therefore, this problem has been completely solved for all forces and all component deflections. (See figure 37.)

In addition to the bolt and frame loads and deflections, there are other gasket bolting problems which are of interest to the designer. Some of these problems will be outlined in the next paragraphs.

It has been seen in previous examples that a gasket deflects much more than its steel retainer. Therefore, if a steel retainer is not sufficiently thick, it may bend and in many cases, permanently yield. This deflection could create a seal leakage problem after only one load application. (See figure 38.)

In some designs it may be necessary to use a large number of bolts to keep a gasket retainer flat. However, this always introduces the danger of overloading and destroying the gasket by preloads supplied by all of these bolts.

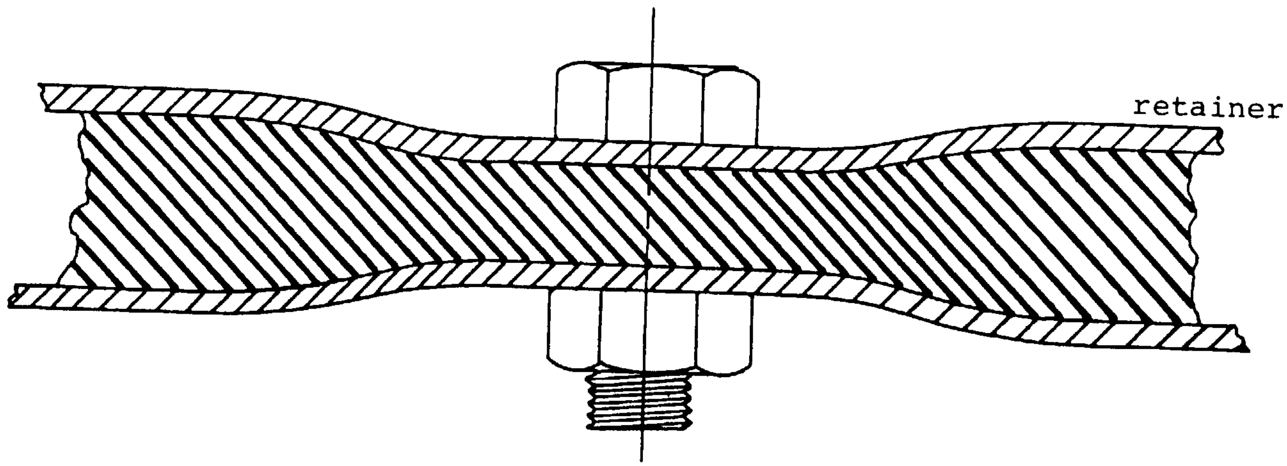


Figure 38 - Bent Gasket Retainer

In some design situations, it may be possible to selectively torque bolts around a gasket flange if the flange is subjected to both a static bending moment and to bolt preloads as shown in figure 39. Assume, for example, that due to the bending moment, the bottom of the flange is in compression and the top in tension. If each bolt is examined independently, it is possible to determine the additional loads due to preload and moment.

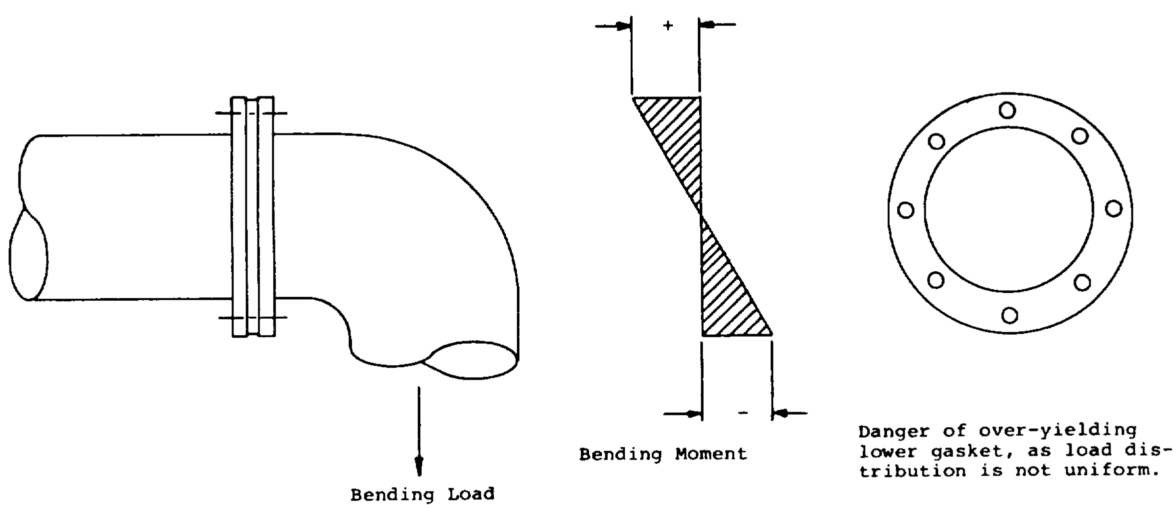


Figure 39 - Non-uniform Loaded Gaskets

In this selective torquing process, no particular bolt will be heavily overloaded. Care must be taken to assure that any of the bolts which end up with very little preload do not rotate loose under any vibratory loads. Locking devices may be the answer to this problem.

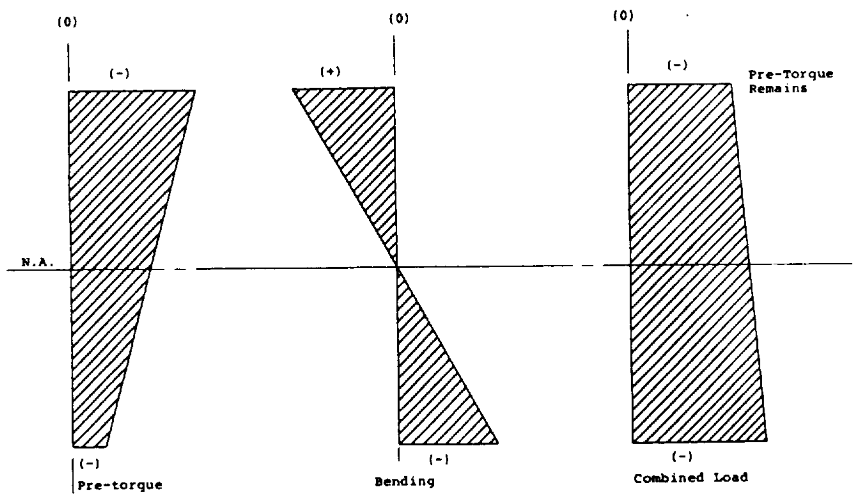


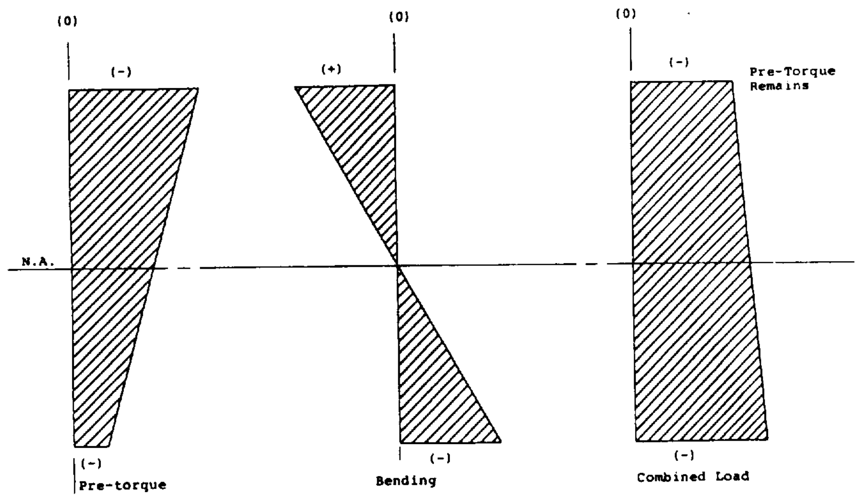
Figure 41 - Relieving Gasket Bending Load

Q.40 How are vibration loads or cyclic loads superimposed on (a) preloads and (b) static loads in a bolted assembly?

A.40 Cycling loads which introduce first tensile added loads to a joint will add to the bolt load and reduce the frame load. As this load reverses the bolt load will reduce and the frame load will increase. Figure 42 shows the time dependent application of (a) the bolt preload, (b) the addition of a static load, then (c) the positive tensile phase of a cyclic load, then (d) the reversed compressive load phase of the cyclic load. Figure 42e shows the entire sequence.

The bolt vibration varies as a function of the load, which starts out with the preload, then increases with more load and goes back to preload, then reduces below preload.

In this selective torquing process, no particular bolt will be heavily overloaded. Care must be taken to assure that any of the bolts which end up with very little preload do not rotate loose under any vibratory loads. Locking devices may be the answer to this problem.



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The bolt vibration varies as a function of the load, which starts out with the preload, then increases with more load and goes back to preload, then reduces below preload.

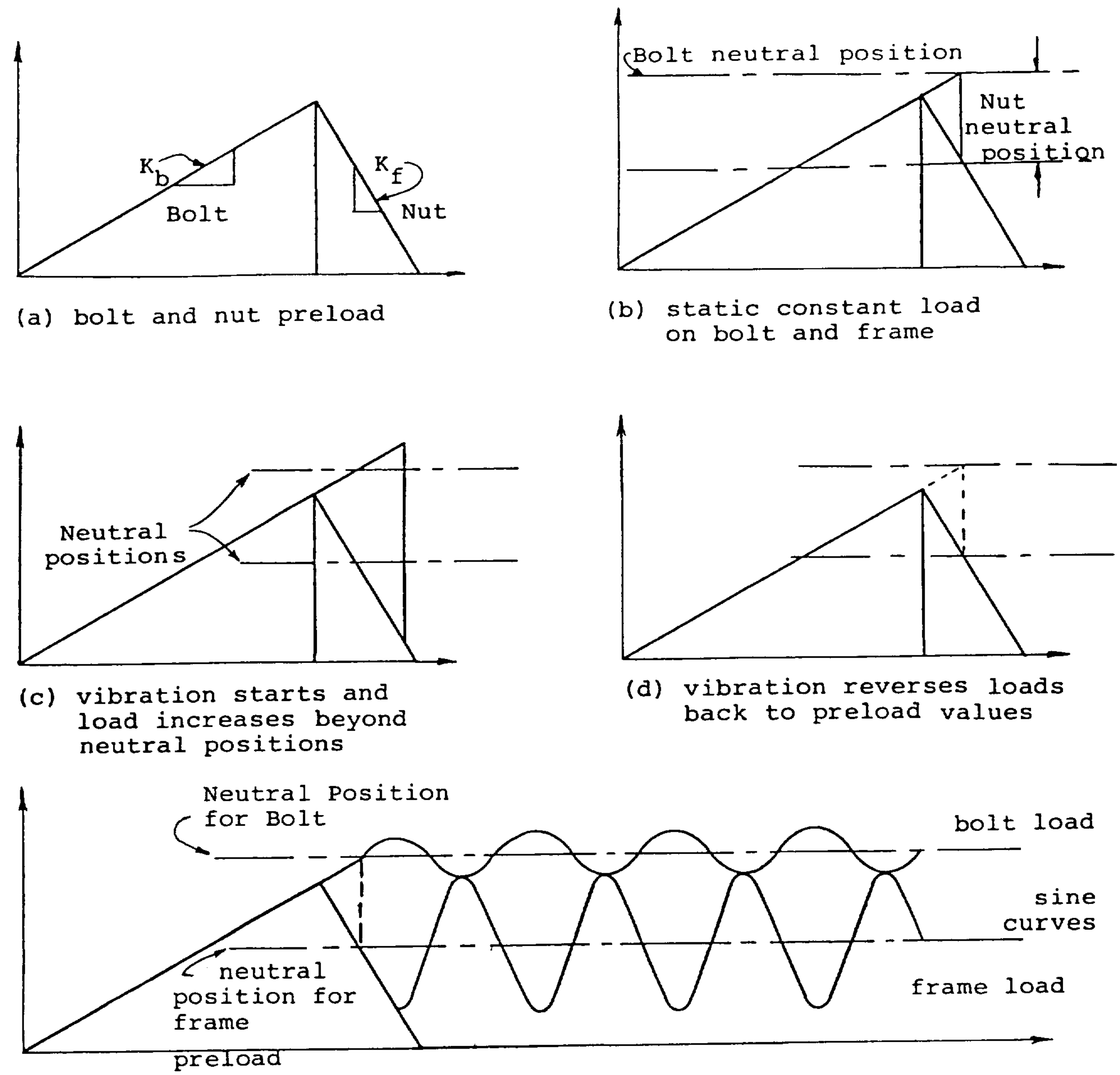

Figure 42 - Steady State Load
First and Subsequent Vibration Loads

Q. 41 How do these vibration loads cause frame gapping?

A. 41 Vibration load-induced gapping is caused when the dynamic inertial forces in the frame exceed the preload force in the frame. Recall that a tension load on a preloaded frame reduces the frame load and increases the bolt load. This is true only up to a point at which the entire frame preload has been removed, after which gapping occurs. As shown in figure 43, the applied load cannot exceed a given preload in the frame, or gapping will occur.

This gapping further induces a non-linear load onto the bolt. The overall joint stiffness constant K also jumps since the effective joint area is reduced to only the bolt area when the frame is unloaded. The bolt will then be loaded well beyond its preload. In frame gapping, the nut and bolt head hang on the frame and support the entire load. When the bolt load reverses, it drives the frame together and there is a sudden strong impact load developed in the whole assembly. This is why selecting the proper preload is so important under dynamic load conditions.

Q. 42 What are some design techniques used to minimize vibration effects in bolted joints? (See Figure 44.)

A. 42 By reducing the stiffness of the frame, more of the vibration inertial load is carried by the bolt. This approach will decrease the magnitude of the cyclic load on the frame and increase the cyclic load on the bolt. (We assume that the bolt load is not critical.) This design method generally reduces the chance of frame gapping.

Alternately, by stiffening the frame, the cyclic load on the bolt is reduced. This decreases the magnitude of fatigue loads on the bolt and increases its fatigue life. However, the chances of frame gapping increase unless a larger bolt preload is applied on the bolt. Gapping is possible since the oscillatory load in the bolt is less. The extra bolt preload will therefore be required to prevent this gapping.

Q.43 So far, only frame tensile loads have been considered. How are frame compression loads analyzed? (See figure 45.)

A.43 A compression load on the frame is obviously not as critical as tension loading on the frame, yet a full understanding is important to frame and bolted joint design. The analysis procedure is similar to the previous load analysis. It may be thought of as simply a reversal of the role of the bolt

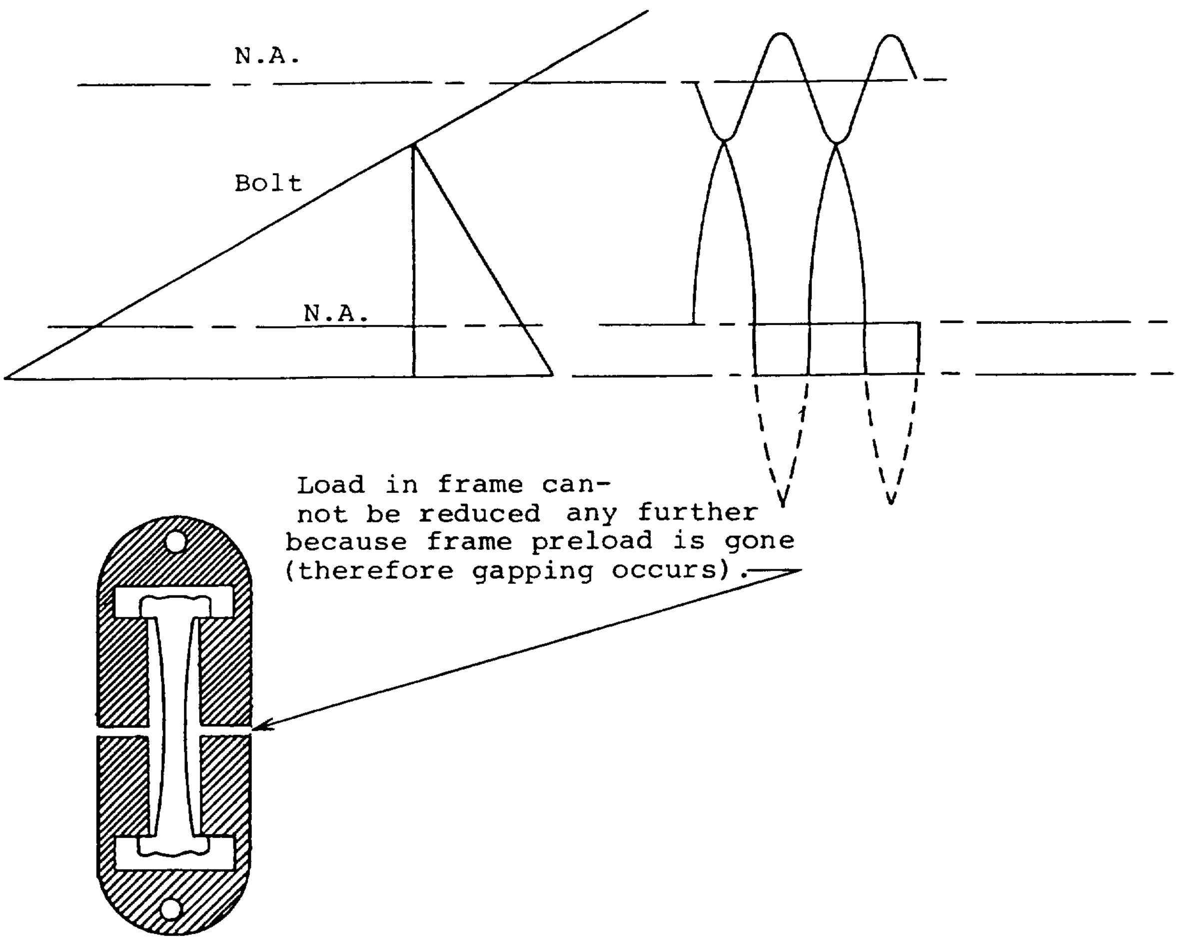

Figure 43 - Vibration Load With Gapping

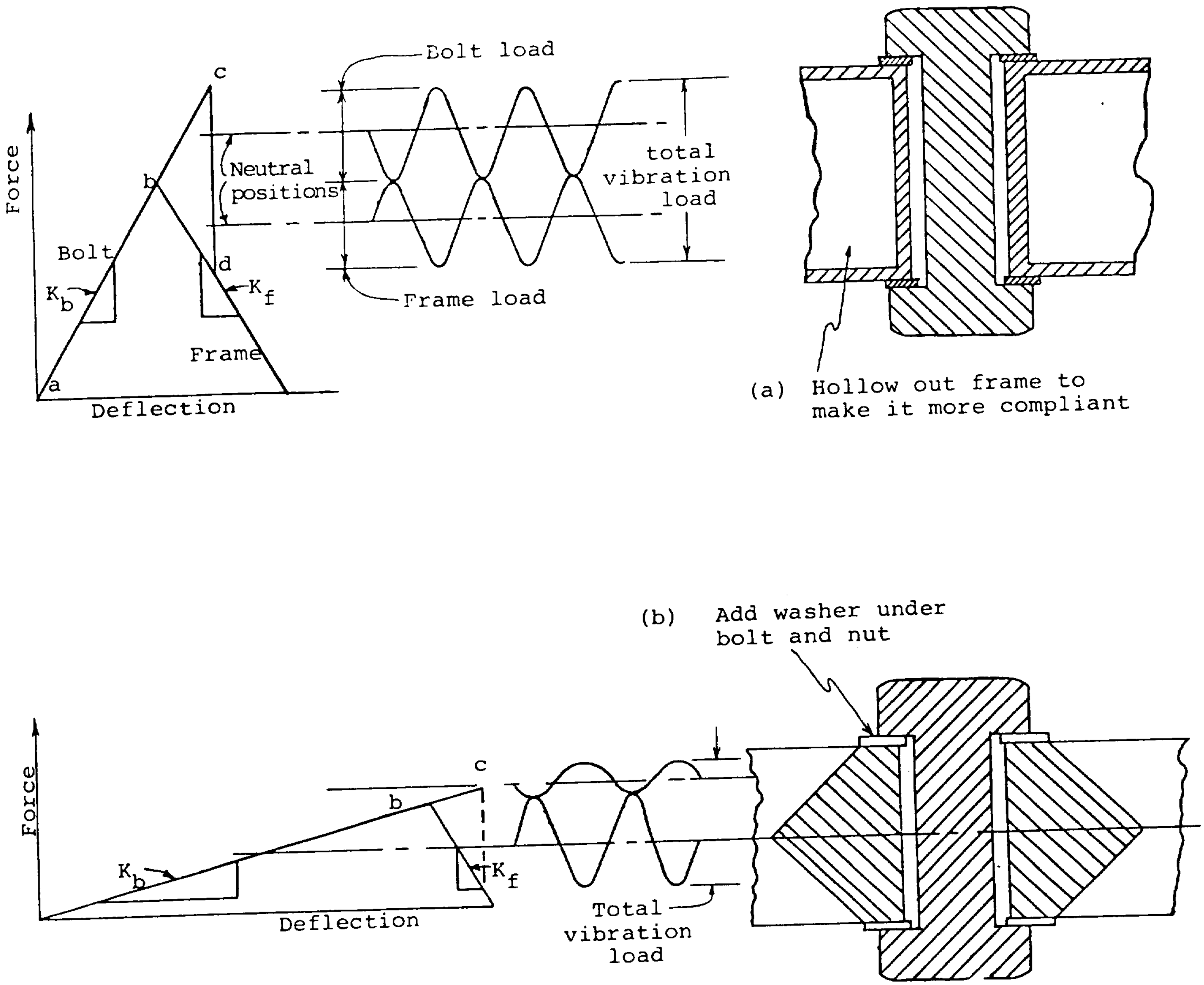

Figure 44- Two Solutions to the Vibration Problem

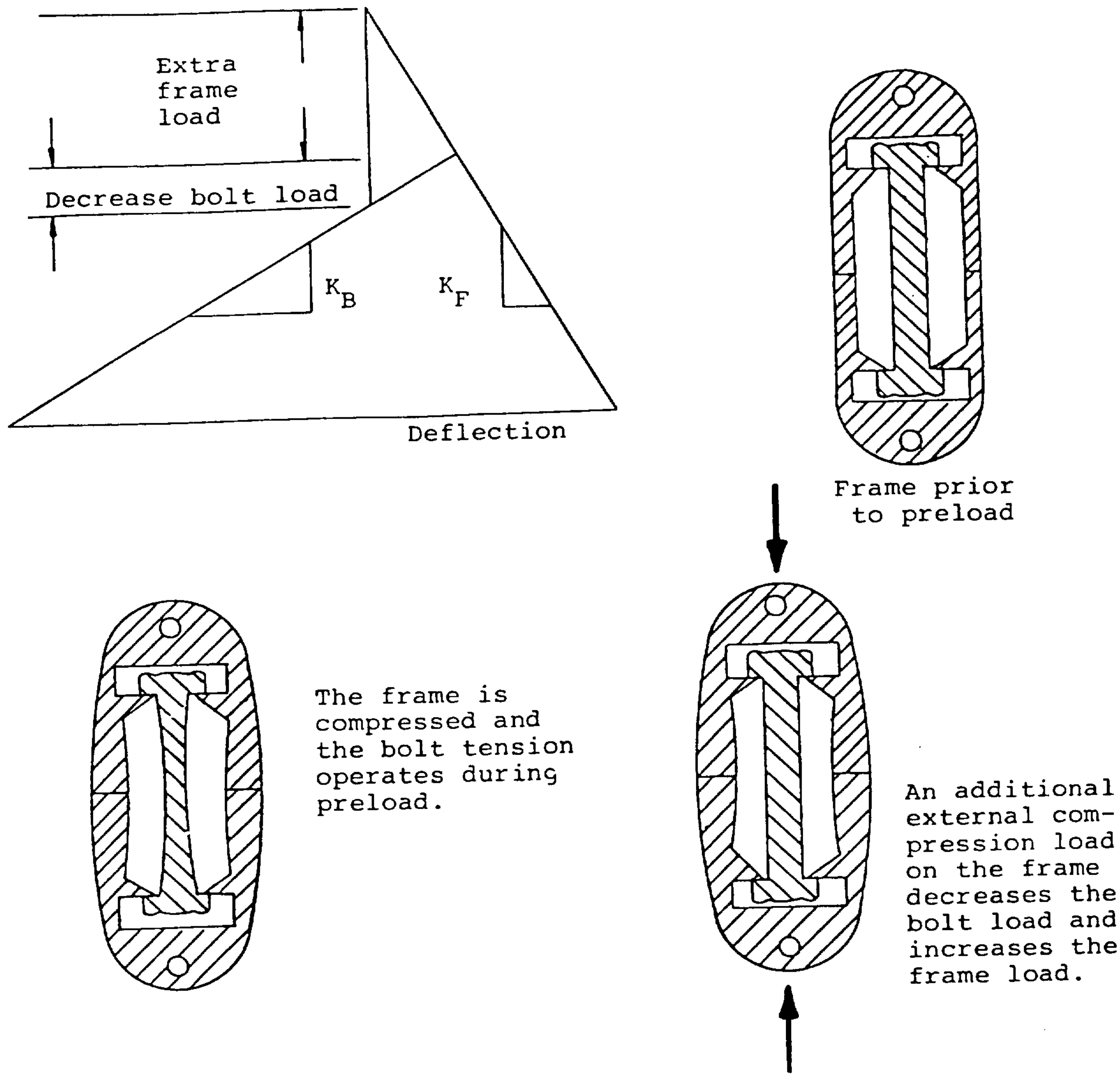

Figure 45 - Compression Loads on Frame

and frame in those analysis equations (i.e., K_b and K_f interchanged).

Q.44 What additional design consideration can assure minimal bolt stresses?

A.44 Design consideration to minimize critical bolt and frame stresses. By adding extra washers and thereby lengthening the bolt's grip and the effective frame length, the K_b and K_f are reduced since this length determines the stiffnesses K_b and K_f (i.e., K_b = EA/L). With these reduced stiffness values, K_b and K_f , the bolts are thus able to absorb more strain energy. The frame's stiffness K_f can also be further decreased by adding compression springs. (See figure 46.)

However, since two structures are generally bolted together, (each with a given mass) care must be taken to be aware of, and to avoid unwanted resonant frequency conditions which would result under vibration loads. This dynamic resonance could further increase vibration loading.

Q.45 What is the comparison between the return yield load capacities of bolts made of low carbon steel and high carbon steel.

A.45 The effect of materials or metallurgy must be discussed first. From figure 14, it was seen that a bolted joint is normally preloaded by applying a specific torque to the bolt. Subsequent additional exterior tensile loads applied to this bolted frame will increase the load on that bolt and will simultaneously decrease the compressive load on the frame.

If the bolt has been preloaded up to its yield point, subsequent additional loads on the system will cause a steel bolt to yield (i.e., the bolt will plastically stretch to a new permanent length). With a high carbon steel bolt, the bolt is not ductile and may not yield. Instead it is likely to rupture (see figure 47) at slightly increased loads. However, a low carbon steel bolt <u>will</u> yield. These bolts will be permanently elongated and will not return to their initial position. Therefore some of the original bolt preload must be lost.

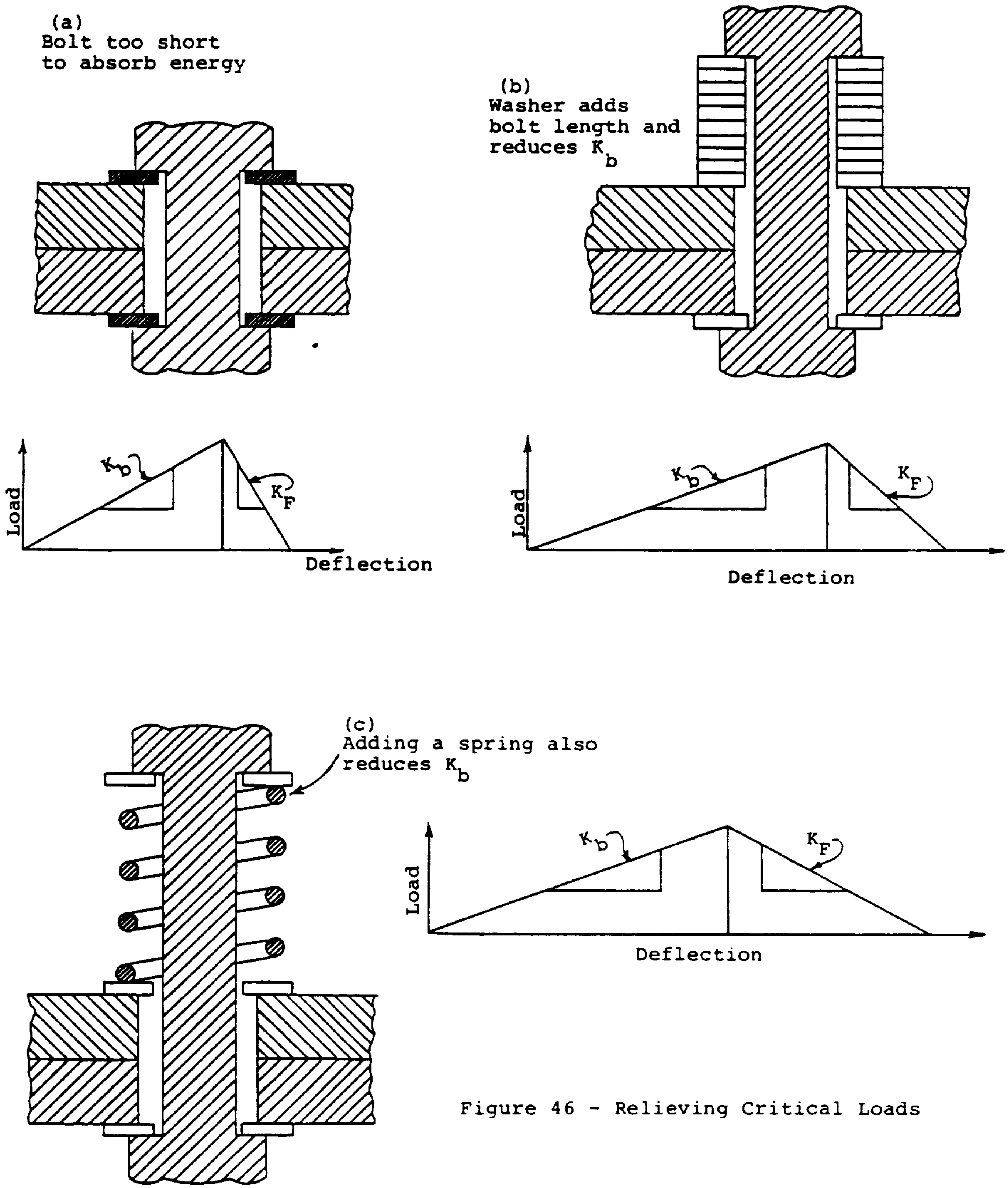

Figure 46 - Relieving Critical Loads

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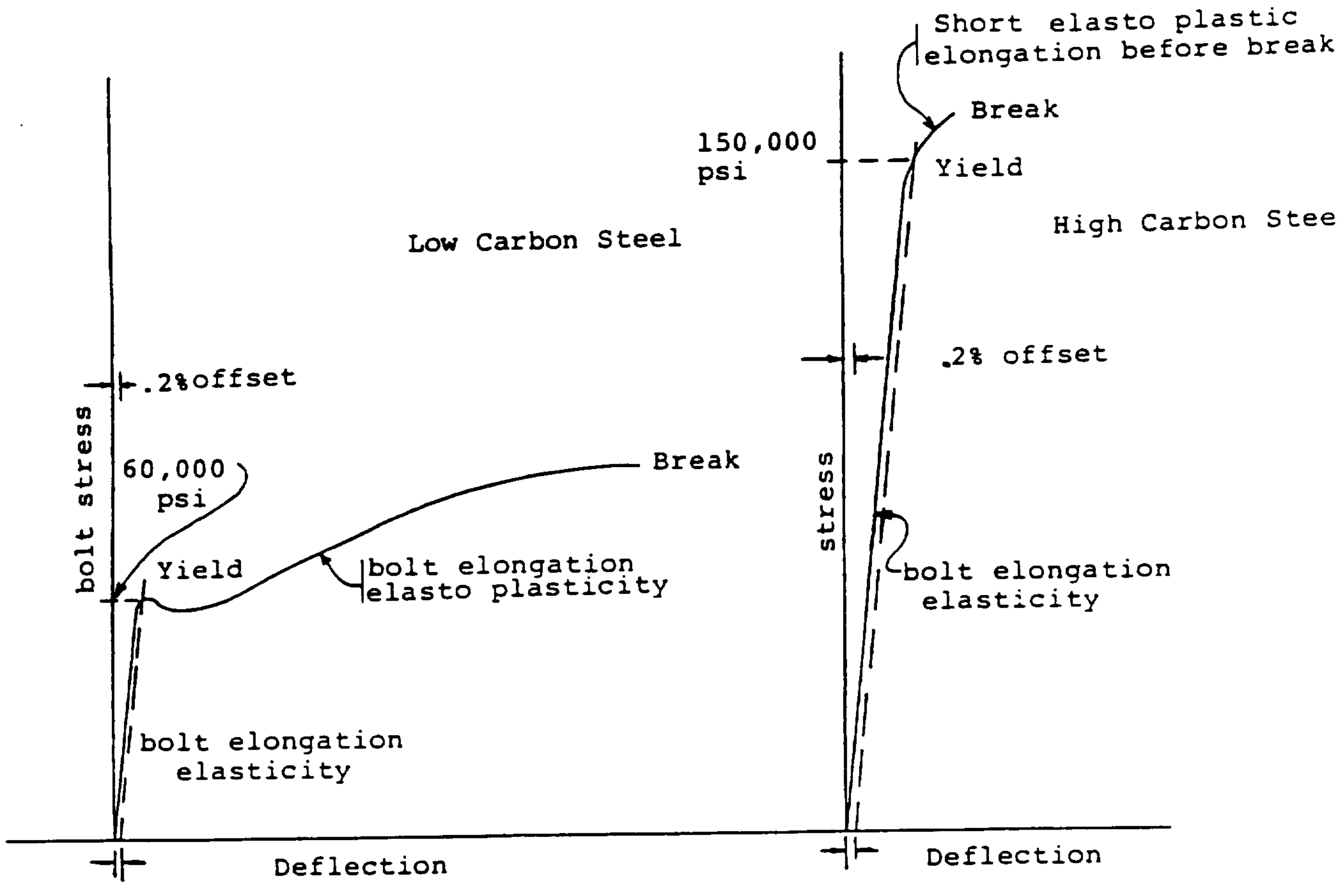


Low carbon steel bolts
stress-strain diagram

High carbon steel bolts
stress-strain diagram

Figure 47- Low and High Carbon Steel Stress-strain diagram

After a bolt has yielded plastically and the assembly is unbolted, it will be observed to have deformed and therefore will not return to its previous loaded condition. This is due to the bolt's permanent set.

If high strength fasteners are used, it is possible to break the bolt joint with little additional load beyond the bolt's yield point (with usually catastrophic results). Therefore, using low carbon construction type bolts, bolt yielding will not generally result in a joint failure when overload occurs. A certain amount of yielding could occur, but since these bolts are generally only used once and discarded, this amount of yielding is often tolerated. In general, it is better (with one use, low carbon bolts) to have a little yielding than to have gapping caused by the use of high strength bolts with insufficient preload torque.

However, designing for bolt yield reduces the bolt's fatigue life and may also cause gapping during loading (see figure 48). It is generally useless to over-stress a bolt during preload as this additional pre-stress is always lost during subsequent application of applied loads.

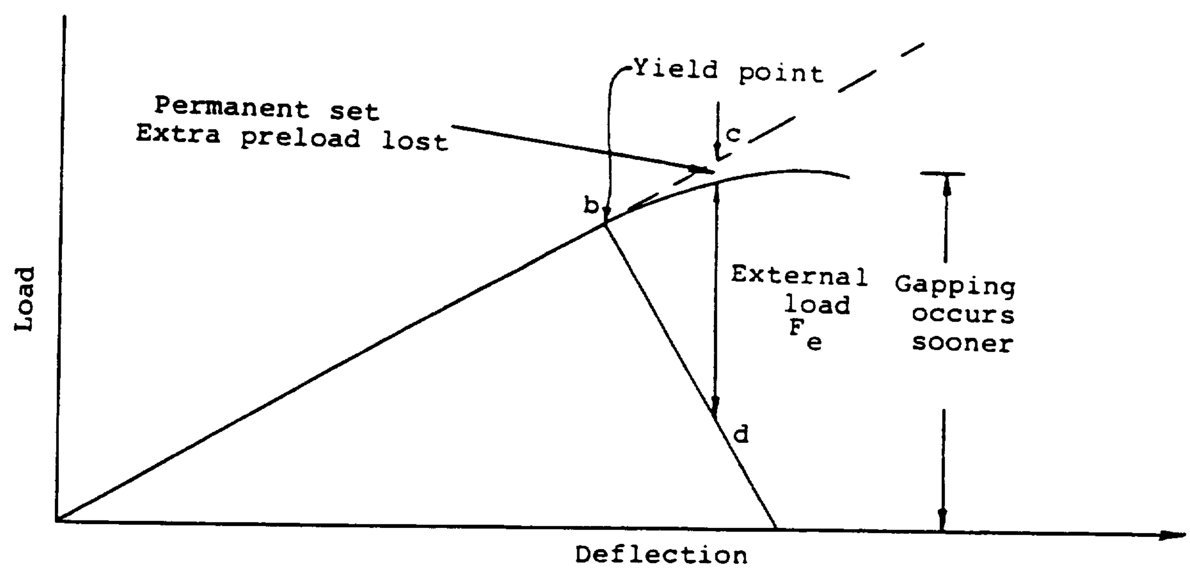


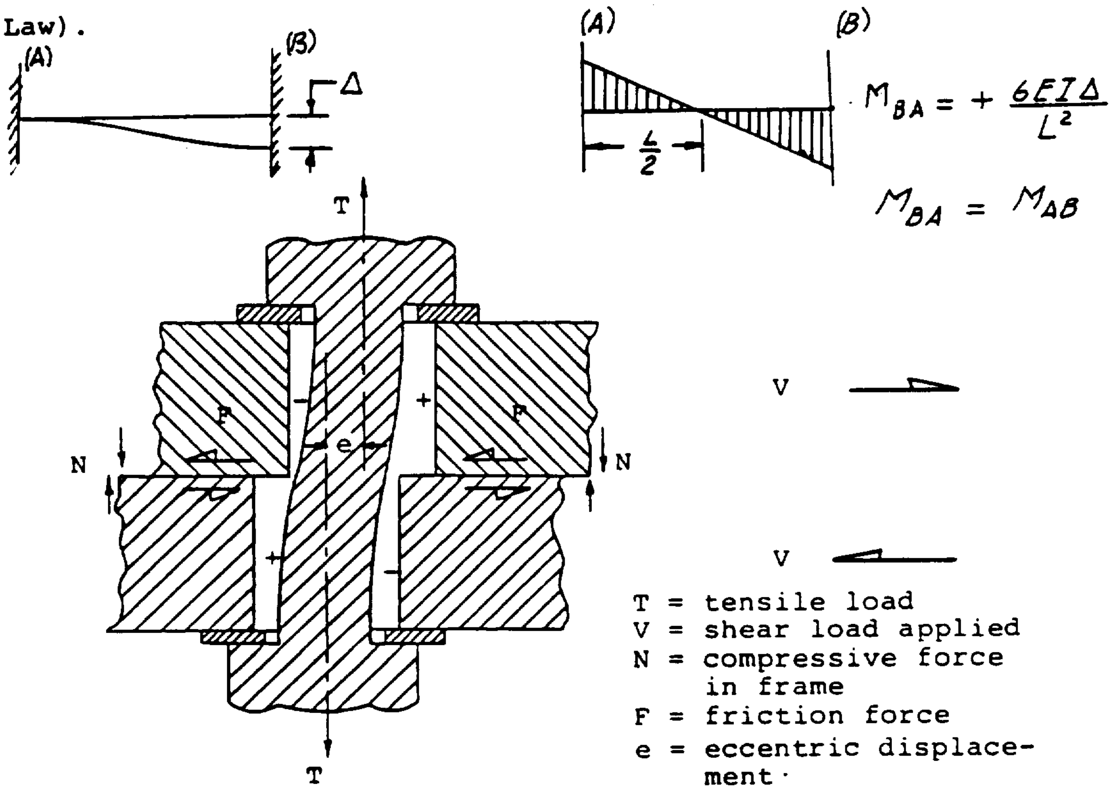
Figure 48 - Bolt After Yield

68N

Q.46 What problems are encountered with bolts which are loaded in shear?

A.46 Bolts designed for shear loading are intended to support loads transverse to their axis.

Shear loads are introduced in a bolted joint when the two plates at that joint are pulled in plane in opposite directions. If the bolt at this joint is preloaded (in tension), the plates will not slip relative to each other. This is provided that the bolt preload is sufficient to react against this external shear by a friction reaction force F (according to Coulomb's Law).



T = tensile load
V = shear load applied
N = compressive force in frame
F = friction force
e = eccentric displacement.

Figure 49 - Bolts in Shear

If, however, the plates should slip, the bolt will take a bent deflected shape as shown in figure 49. Since the bolt was preloaded in tension, this tension (marked "T" in figure 49) will cause a further flexural stress in the bolt due to the added moment Te .

There is then always a danger of bolt fracture if a shear bolt is preloaded to a high percentage of its yield load. The bolt could yield in flexural stress and consequently lose some of its preload. There is also a danger of frame gapping with even further yielding of the frame.

If axial vibration loads accompany a shear load V, there will be a danger that these further loads will remove all of a bolt's preload and thereby reduce the friction restraining forces, resulting in continuous impacting of the bolt on the frame structure.

Q.47 Why is it difficult to define yielding in bolts?

A.47 For a specimen under tension or compression, yield is generally defined at the break in the linear portion of the stress-deflection curve (see figure 50) at a certain offset (usually .2% of the gage length). If the gage is 1.0 in long, then the offset = .002 inches.

All of the previous calculation curves referred to the yield load of the entire bolt while true yield stress will vary from point to point within the bolt.

Bolt stresses could therefore reach their yield points along the entire bolt length as in figure 51, or only locally about some point. A small locally yielded point in the bolt could then go unnoticed during torquing or during use. Overall bolt yielding is measured along the entire length and not examined locally.

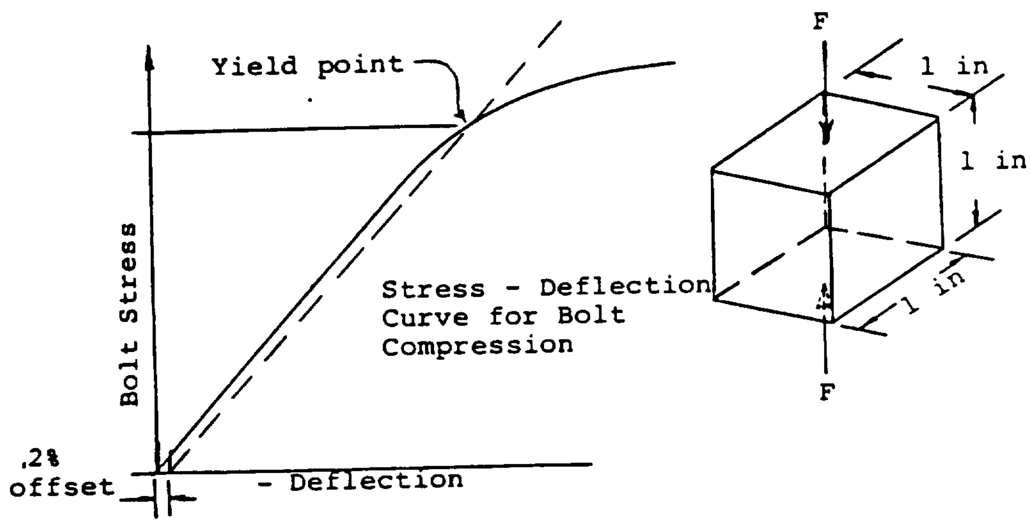


Figure 50 - Definition of Yield Point

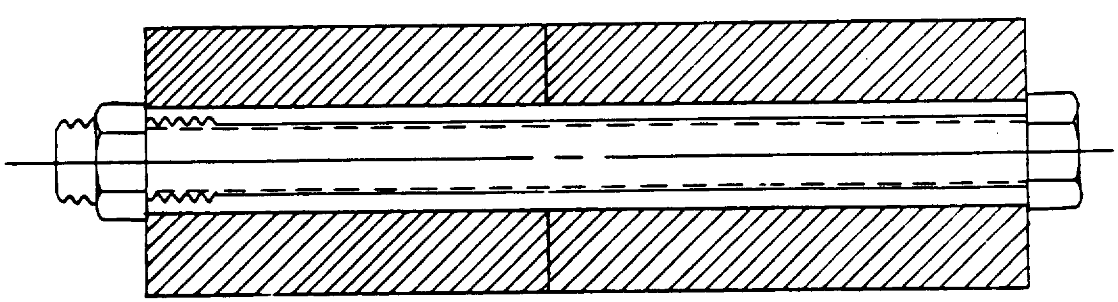


Figure 51 - Entire Bolt in Yield

SECTION IV

Verification of Axial Preload Procedures Used

Over the past ten years NASA GSFC personnel have conducted various tests to evaluate the reliability of the procedures recommended by this manual.

Q. 48 What are some of the tests performed at NASA Goddard Space Flight Center to verify the previous clamping and loading methods?

A. 48 A series of bolt torque and preload tests were performed.

These test series were conducted by using a fixed size bolt set and by varying the plate bearing areas.

Summary of the NASA GSFC Test Results:

(1) With low preloads gapping began to appear, even at 50% of preload (if full stress up to yield is assumed).

(2) When gapping occurs, the joint spring constant changes suddenly causing a severe impact load to the structure.

(3) With too much preload torque yield occurs almost immediately upon application of an extreme external load. This yield then causes a corresponding loss in preload.

(4) Failure generally occurs when the ratio $R = K_B/K_F = 1/2$ with an external load over 1.0 times yield load.

(5) With an increase in "R" ($R = K_B/K_F$) from 1/4 to 1/2, there is less gapping but more bolt yield and quicker bolt failure.

(6)

(7) Pre-torquing up to full yield causes an eventual loss of pre-torque. A design should not go that high in torque just to lose it with the corresponding increased danger of failure. Figures 52 and 53 show the load-deflection relations for some of the tests conducted.

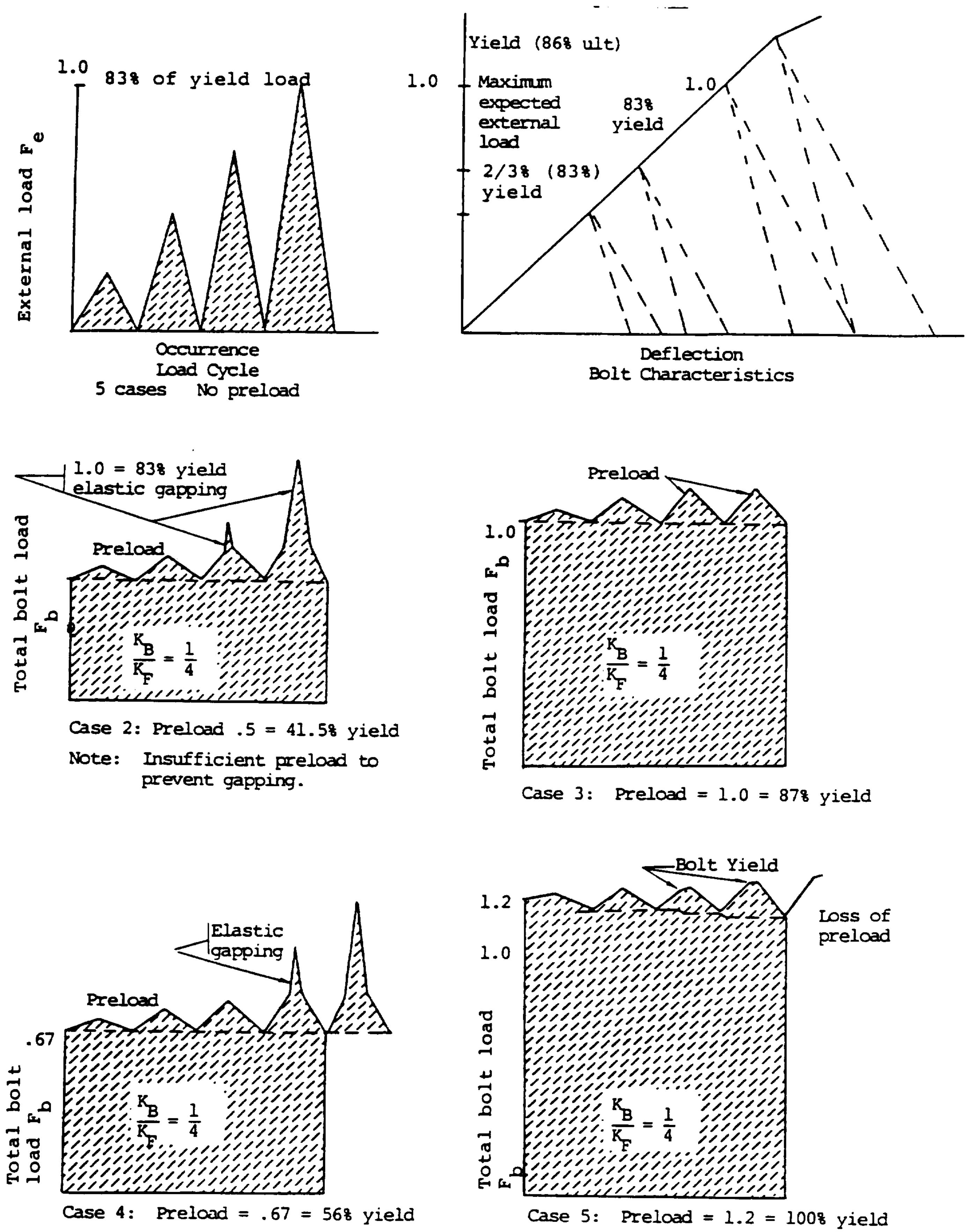


Figure 52 - Gapping and Yielding of Bolted Systems for 5 cases

[redacted] 73

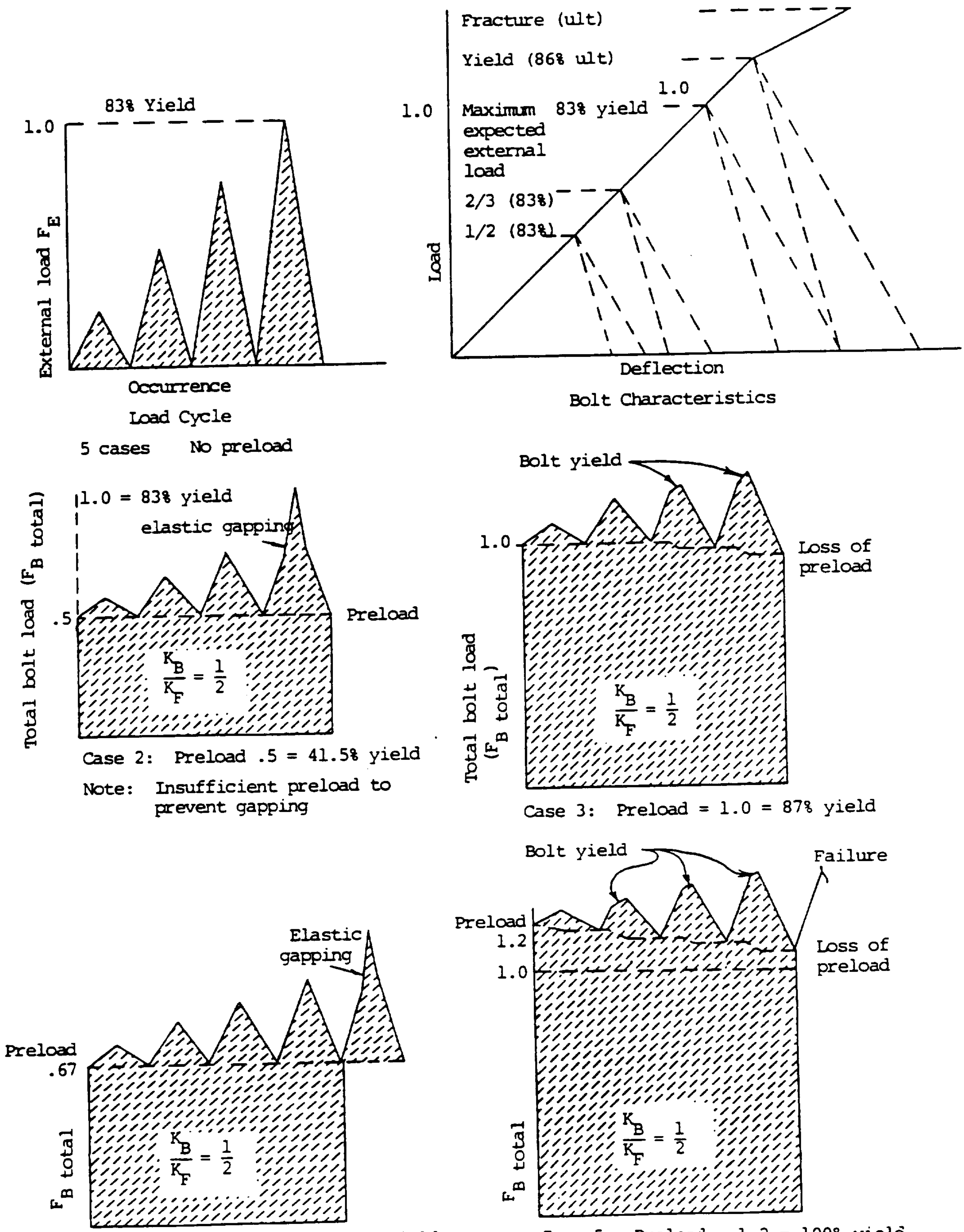


Figure 53 - Gapping, Yielding and Failure of Bolted System

Q. 49 How are the results of these tests and the previous Section III analysis used to select a joint preload method?

A. 49 The results of this manual's methods have been presented in a convenient and graphical form (figure 54) for easier use, which will be described below. Recall that the ratio R was the ratio of bolt to frame stiffness, which is a function of the part area, i.e.,

$$R = \frac{K_B}{K_F} = \frac{\left(\frac{A_b E_b}{L_b}\right)}{\left(\frac{A_f E_f}{L_f}\right)}$$

K_B = bolt stiffness

K_F = frame stiffness

Referring to figure 54 and equation A5, the additional bolt load due to an external load is found from:

$$F_{B_1} = \text{(bolt additional load)} = \frac{R}{R+1} \text{ (external force } F_e)$$

Therefore the total bolt load is :

$$F_{B(\text{final})} = F_{B(\text{preload})} + F_{\text{External}} \left(\frac{R}{R+1}\right)$$

Yet

$$\frac{R}{R+1} = \frac{1}{1+1/R}$$

Therefore, the final bolt load is:

$$F_{B(\text{final})} = F_{B(\text{preload})} + F_{(\text{External})} \left(\frac{1}{1+1/R}\right)$$

The test series conducted proved the general design philosophy provided in Section III. Then for the bolted joint, this design philosophy indicates that:

25

(1) One should design a joint to yield under its full expected yield load.

(2) Preload the bolt only up to a limited percentage of its yield load (figure 54).

(3) Use the graph's ordinate (on the left of figure 54) to determine that additional load capacity of F_E which is available after the bolt is preloaded and for which either the bolt reaches its yield or for which the joint gapping would occur.

Examples of how to use the nomograph in Figure 54

Select a stiffness ratio. For example, select $R = 4/1$ for a soft frame representing a gasketed joint, or $R = 1/8$ for a stiff solid frame.

Assume, for example, that $R = 1/4$, and that it is desired to find that percentage of a bolt's yield load which the bolt should be torqued up to in order to assure that 100% of the bolt's yield capacity is used for a given external load, and at the same time to prevent gapping. This value would then be an optimum design torque for bolted joints.

To find this torque, enter figure 54 ordinate along the top heavy line (which corresponds to 100% of the yield load experienced during a joint's use). Move across this line to the right to the point where the $R = 1/4$ diagonal line intersects the top horizontal line. Then read vertically downward to the point on the abscissa to where the preload required is 80%. This is the torque required for the joint.

In a second example, suppose a gasket seal is being torqued. In the case of this gasket, assume $R = 4/1$. If the bolt is preloaded to 80% of its yield, then using figure 54, find the maximum extreme external load that one could use for this system.

The stiffness ratio is: $R = \dfrac{K_B}{K_F} \dfrac{(\frac{AE}{L})_B}{(\frac{AE}{L})_F}$ where K_B = bolt stiffness, K_F = frame

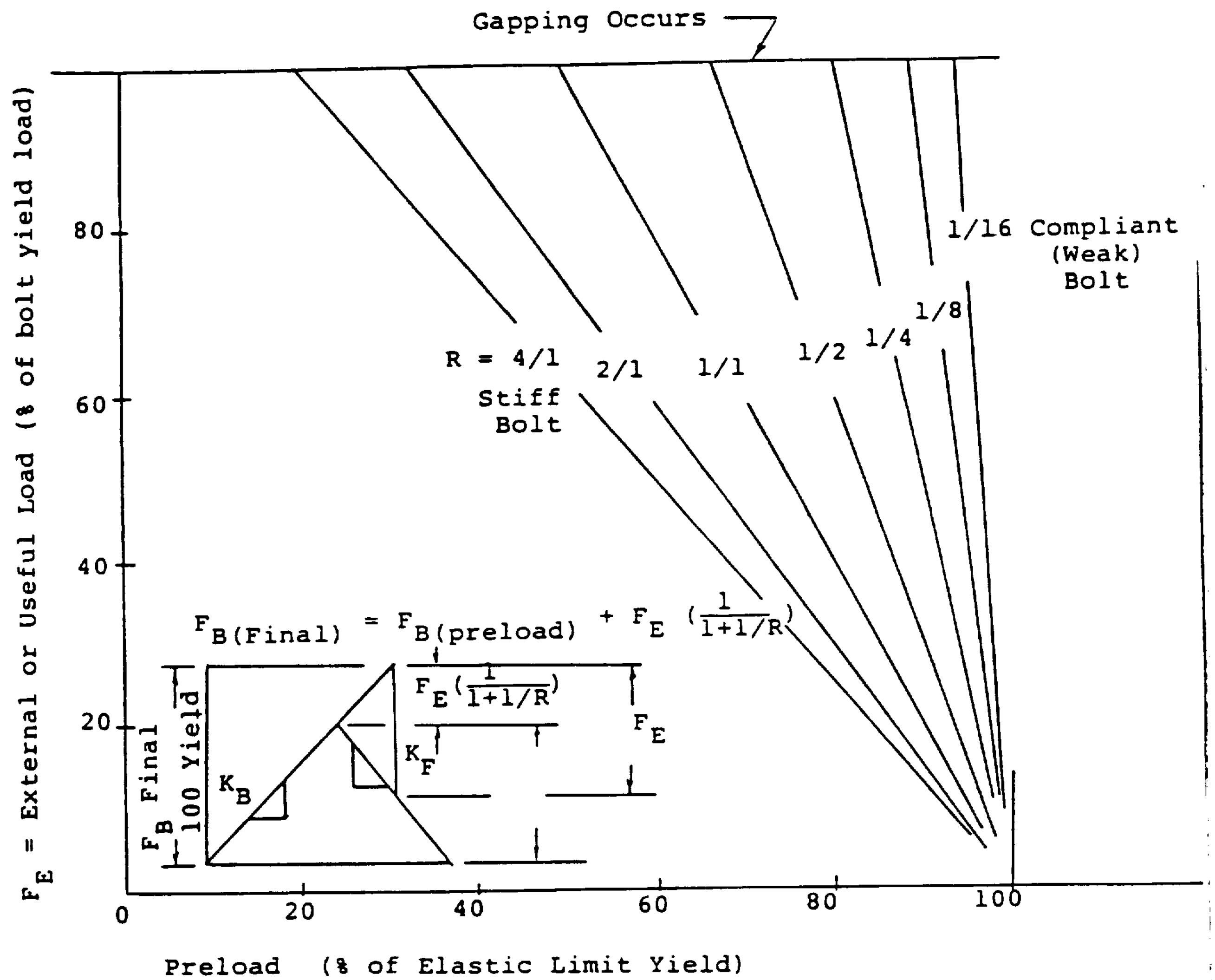


Figure 54 - Bolt Capacity vs Preload

This is approximately 23% of bolt yield. This is found by entering the figure 54 at the bottom with a torque of 80% of yield. Go up vertically to the R = 4/1 diagonal line then across to find F_E = 23%. Figures 55-61 show these same results obtained in terms of the bolt load diagrams previously used.

If a gasket joint with an R = (2/1) is preloaded to 20% of yield, it will gap before an external load of 100% of yield can be applied. Observe this from figure 54 by entering the graph at a torque of 20%. Go vertically to R = 2/1. What F_E is found? Is this value above the gapping line?

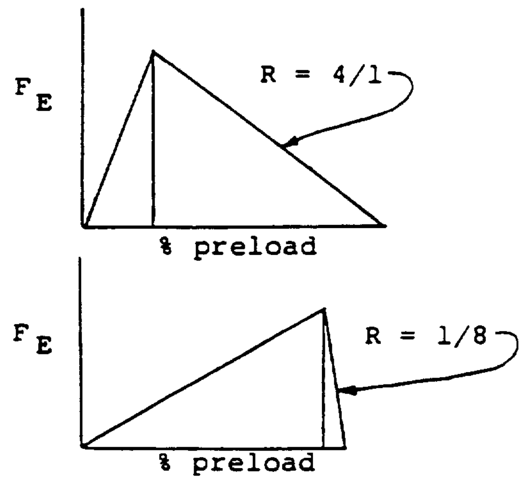


Figure 55

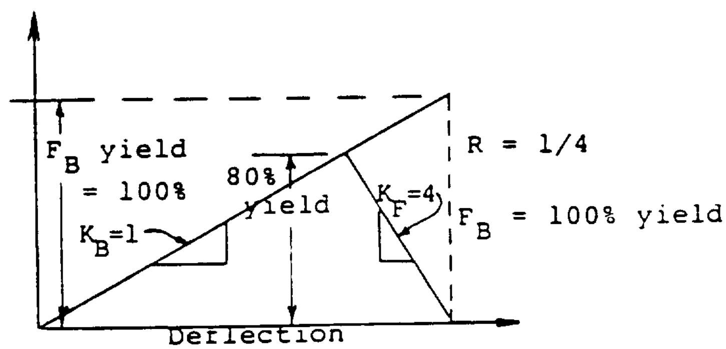


Figure 56

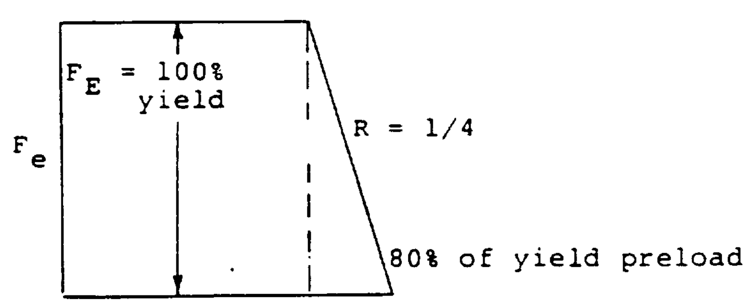


Figure 57

Examples of Using Nomograph in Figure 54

Figure 58

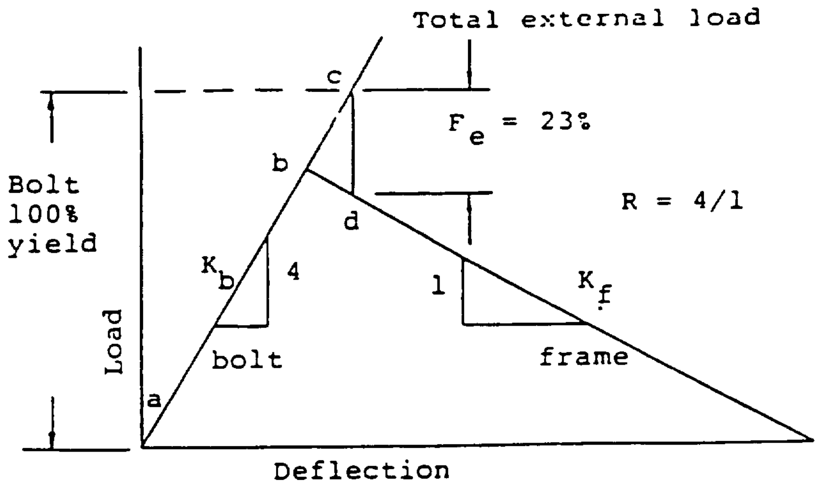


Figure 59

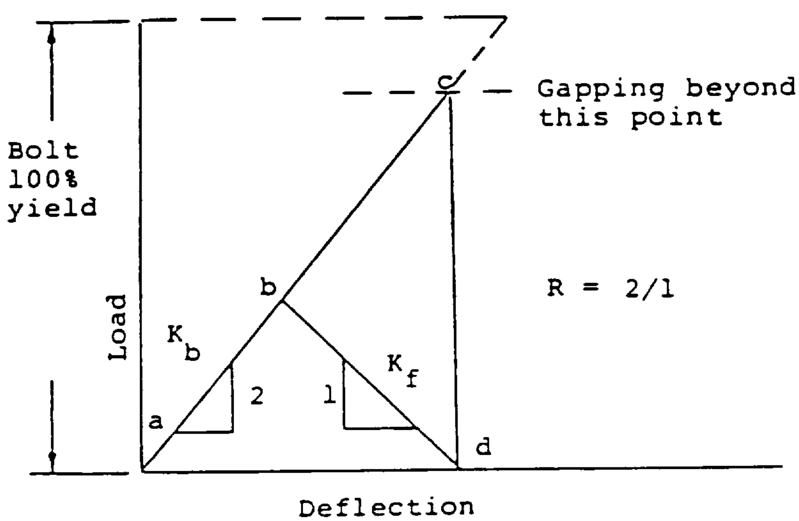
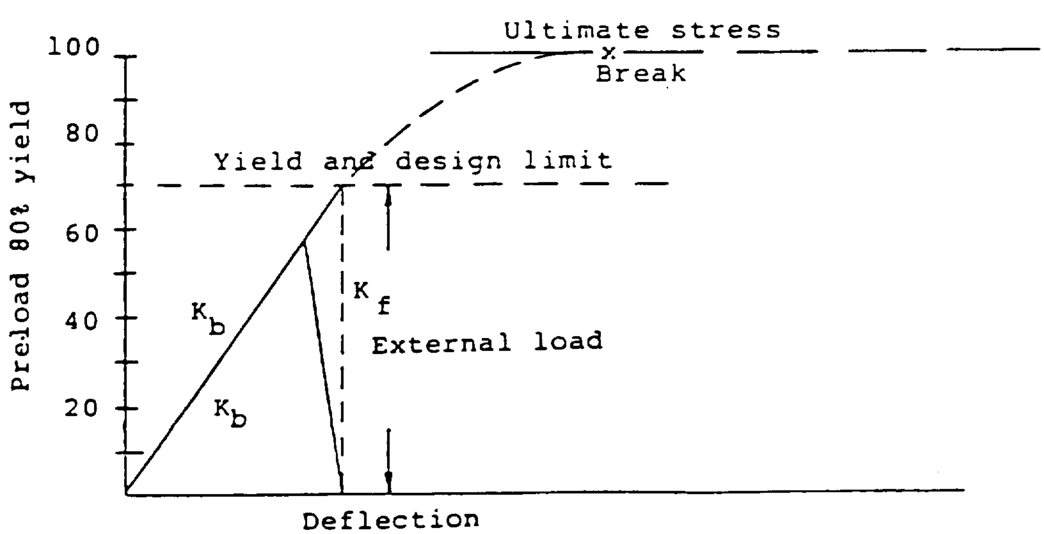


Figure 60



Examples of Using Nomograph in Figure 54

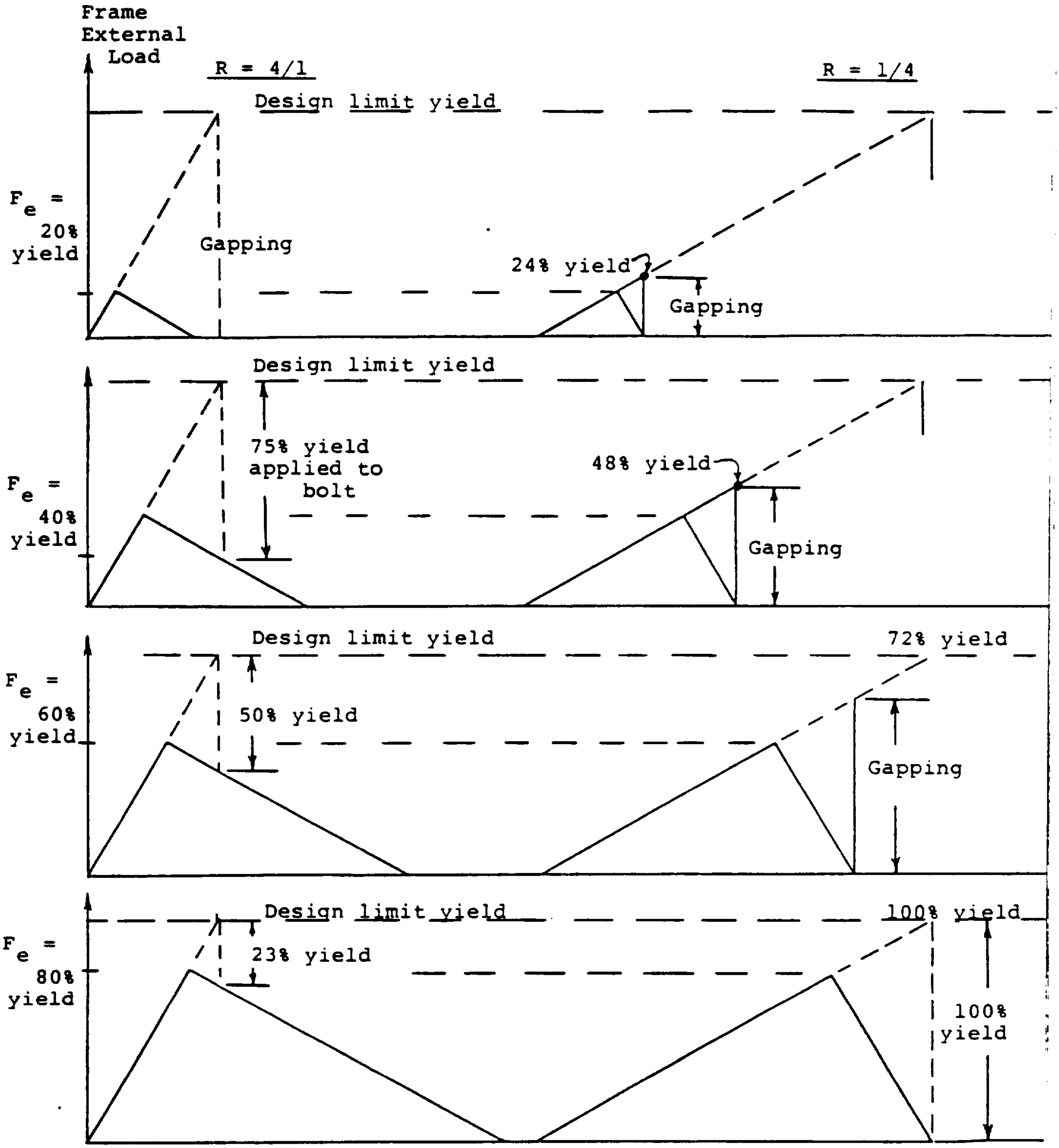


Figure 61 - Summary of Various Preloads and Gapping

SECTION V

Practical Design Considerations for Shear Loaded Joints

In the previous sections of this manual, it was described how bolted joints or frames would yield both in axial tension and in compression.

Bolted joints must generally be investigated for four types of shear failure: fastener shear, bearing, tearout, and shear tension. We have examined only joint tension and compression in the previous sections.

The true stress distribution for all modes of shear failure is extremely complex and the designer is referred to advanced texts on elasticity or finite element analysis methods. In each of these shear load cases it is customary to assume a uniform or average stress distribution in a fastener or bolted plates and to rely upon test results to determine the allowable average stress. These allowable stresses are a function of the joint geometry as well as of the material. Figure 62 depicts the four dominant modes of shear joint failure with corresponding average stress equations.

Q. 50 What are some of the stress combinations we should examine in a shear connection?

A. 50 A designer must examine the shear load in bolts and the combined shear stress induced in these bolts.

82 N [redacted]

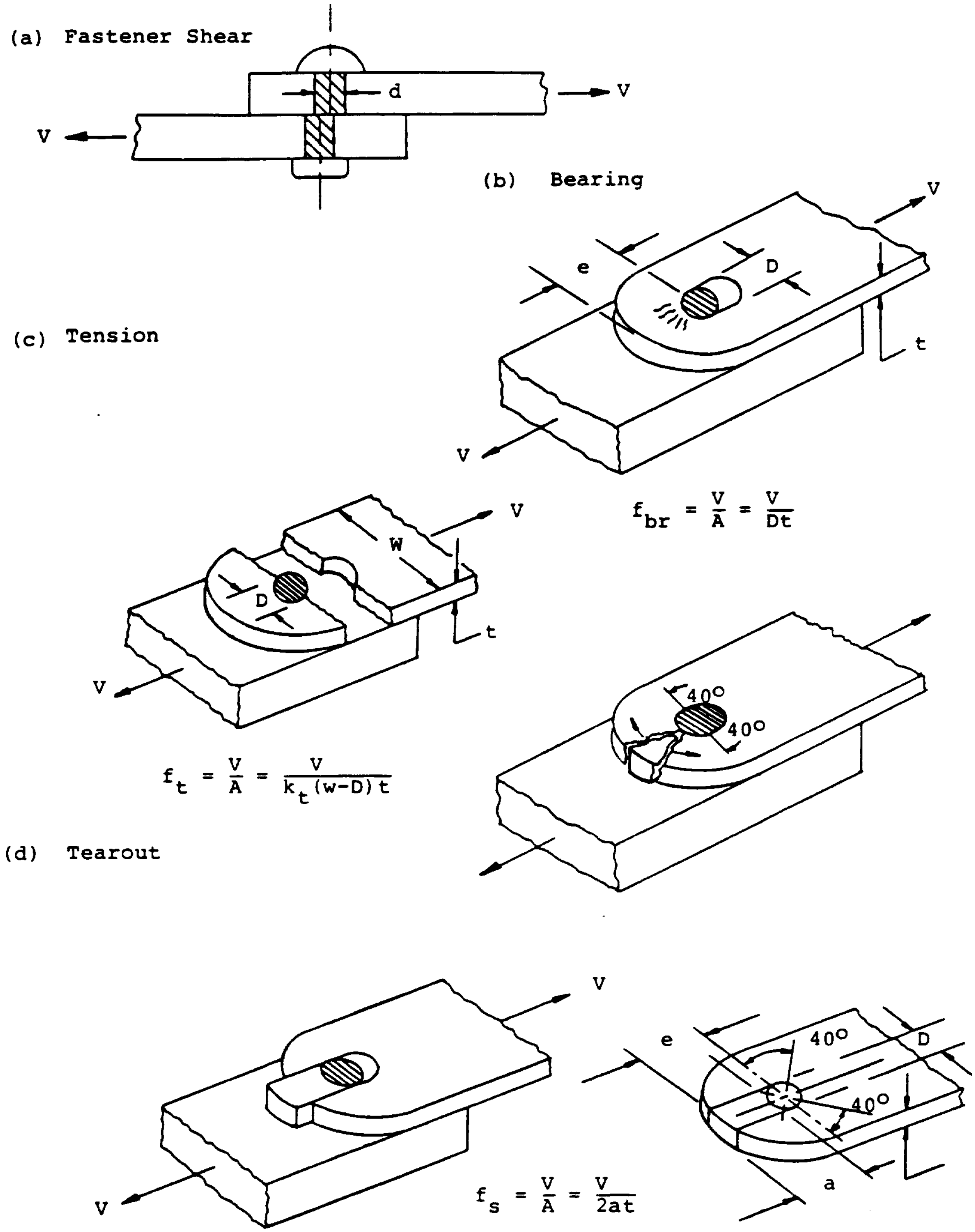


Figure 62 - Types of Shear Failure

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The shear stress under a load V in shear = (V/A) k, where k is a shear factor of 4/3 for bolts, and A is the area of the solid diameter or that part of the bolt in shear. (See figure 63.)

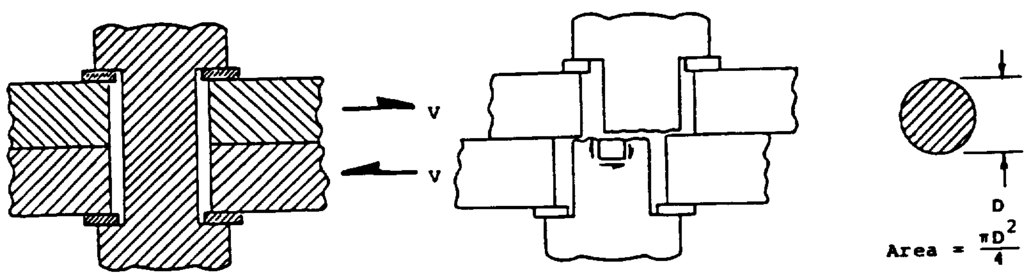


Figure 63 - Shear stress under load

This shear stress must be added by vector addition to the predicted tensile or compressive stress. One method of vector addition is through Mohr's Circle, as discussed in Appendix A3, and as shown in figures 64 and 65.

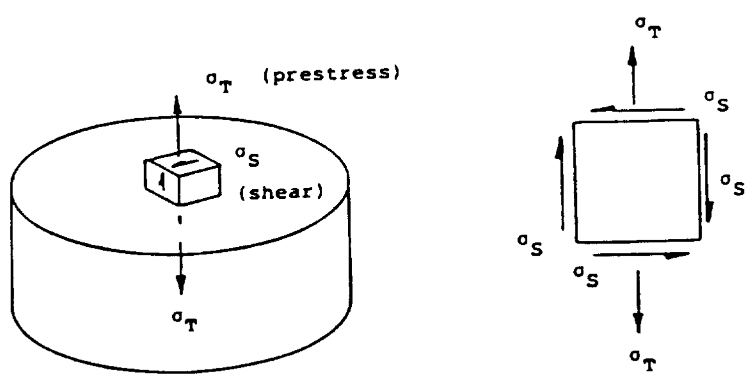


Figure 64 - An Element under Shear and Tension

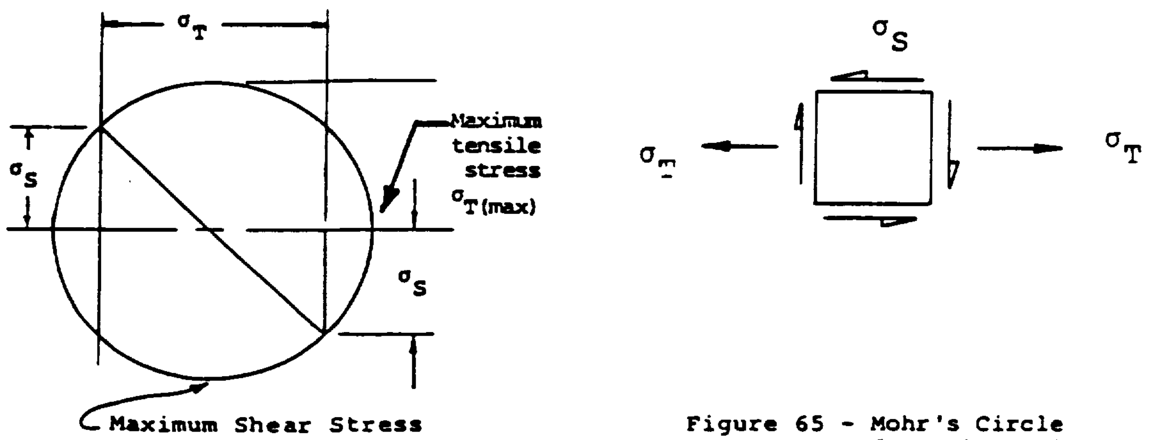


Figure 65 - Mohr's Circle for Calculating Maximum Shear Stress

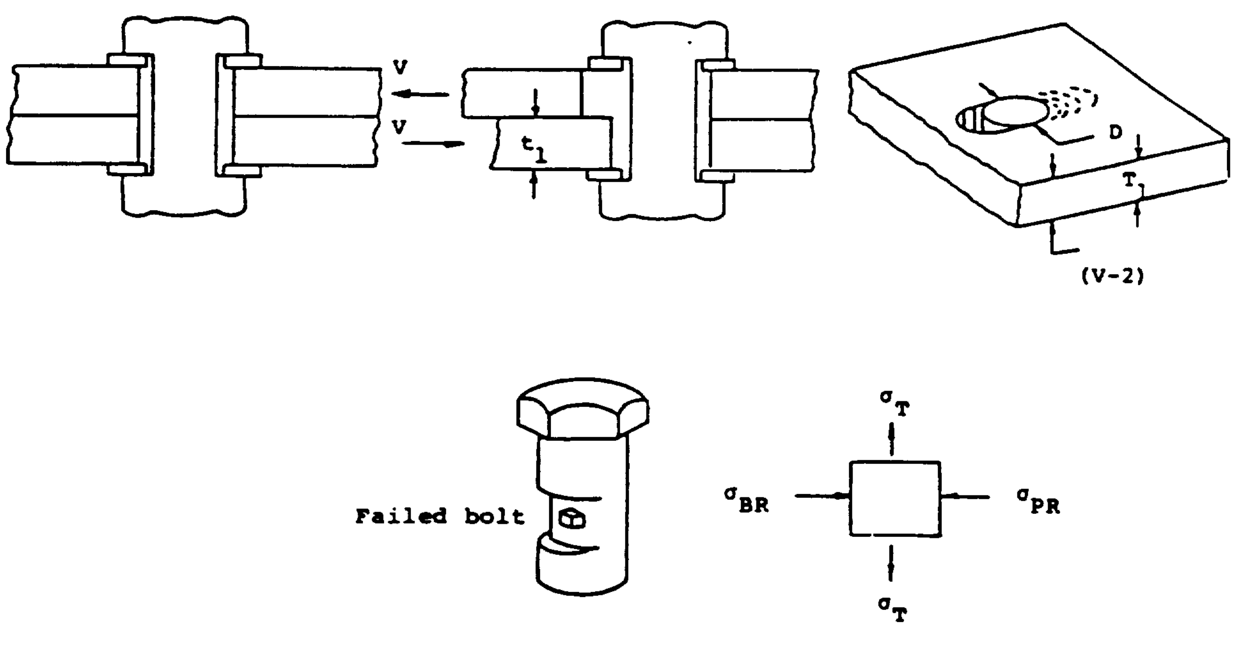


Figure 66 - Bearing Stress in Bolt

This type of shear failure is most evident in:

- thick plate and small bolt diameter designs.
- where steel plates and aluminum bolts are used.
- where high strength plates and low carbon steel bolts are used.

While bolts are generally preloaded to a large fraction of their yield strength, additional shear load stresses (due to applied service loads) must always be added vectorially to the preload stresses. Therefore, bolts may fail due to these combined shear and tension loads. Thus, it is important that every loading case be analyzed to find the magnitude of the shear stresses.

The second type of loading condition (shown in figure 66) is a bearing load (where the plates "bear out" on the shank of the bolt).

The one serious design problem often encountered in a shear joint is that the bolt is threaded in its would-be bearing zone. This is an extremely bad design practice and must always be avoided, since a thread point load failure in the bolt and the plate surface will occur in that situation.

Even though the bolt preload is in tension and the bearing stress is in compression, the bolt could fail under these conditions in combined shear along a plane oriented at approximately 45° (see figure 67). This is explained by the Mohr's Circle stress calculations (see Appendix A-3).

If a plate is in bearing and since $R = K_b/K_f$ is usually 4 or higher for a good design, the bearing stress in the plate will usually be larger than the compressive stress due to the applied preload developed under the head of the bolt or washer. (See figure 68 for Mohr's Circle of stresses in the plate).

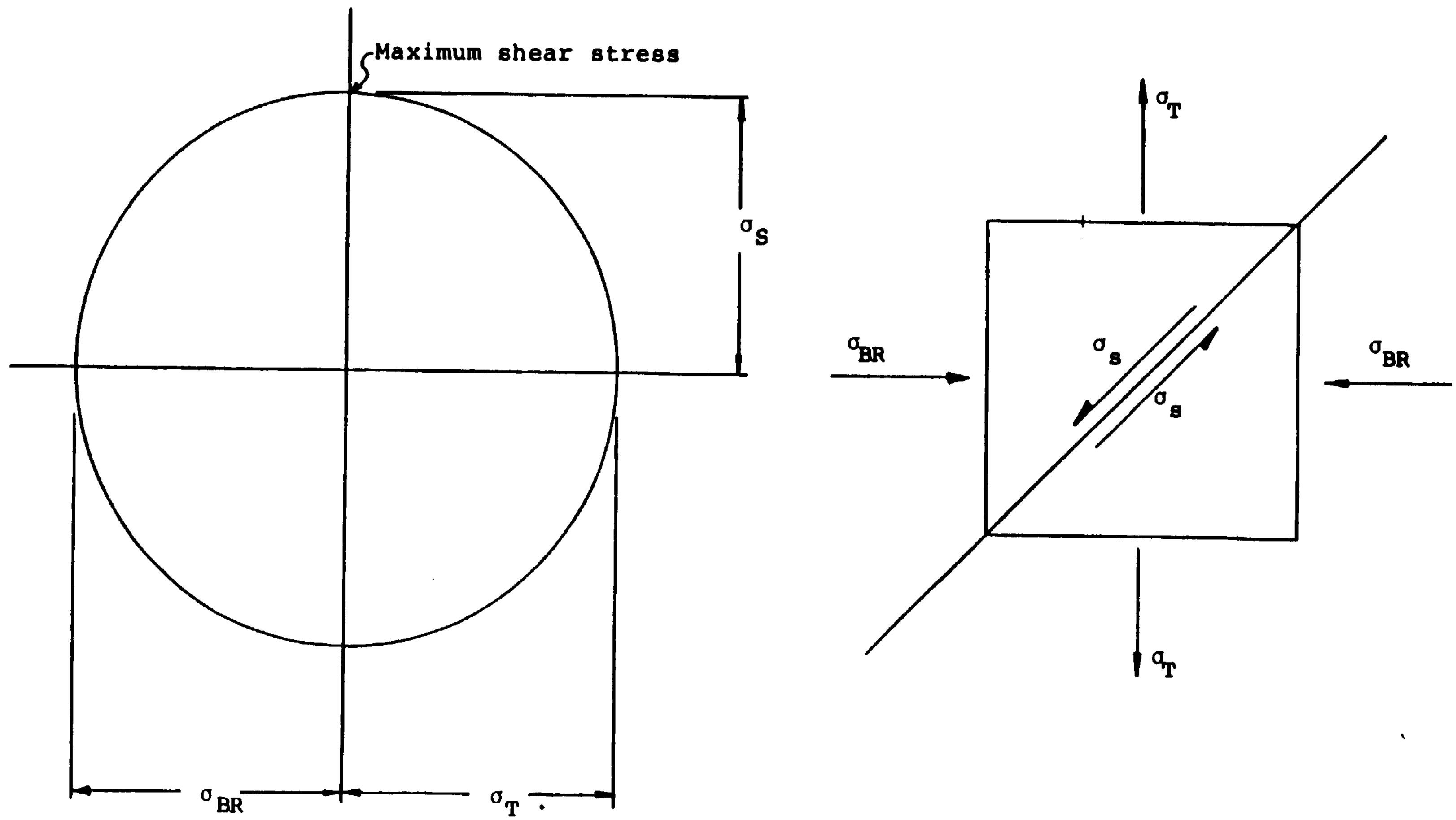


Figure 67 - Mohr's Circle for combined shear

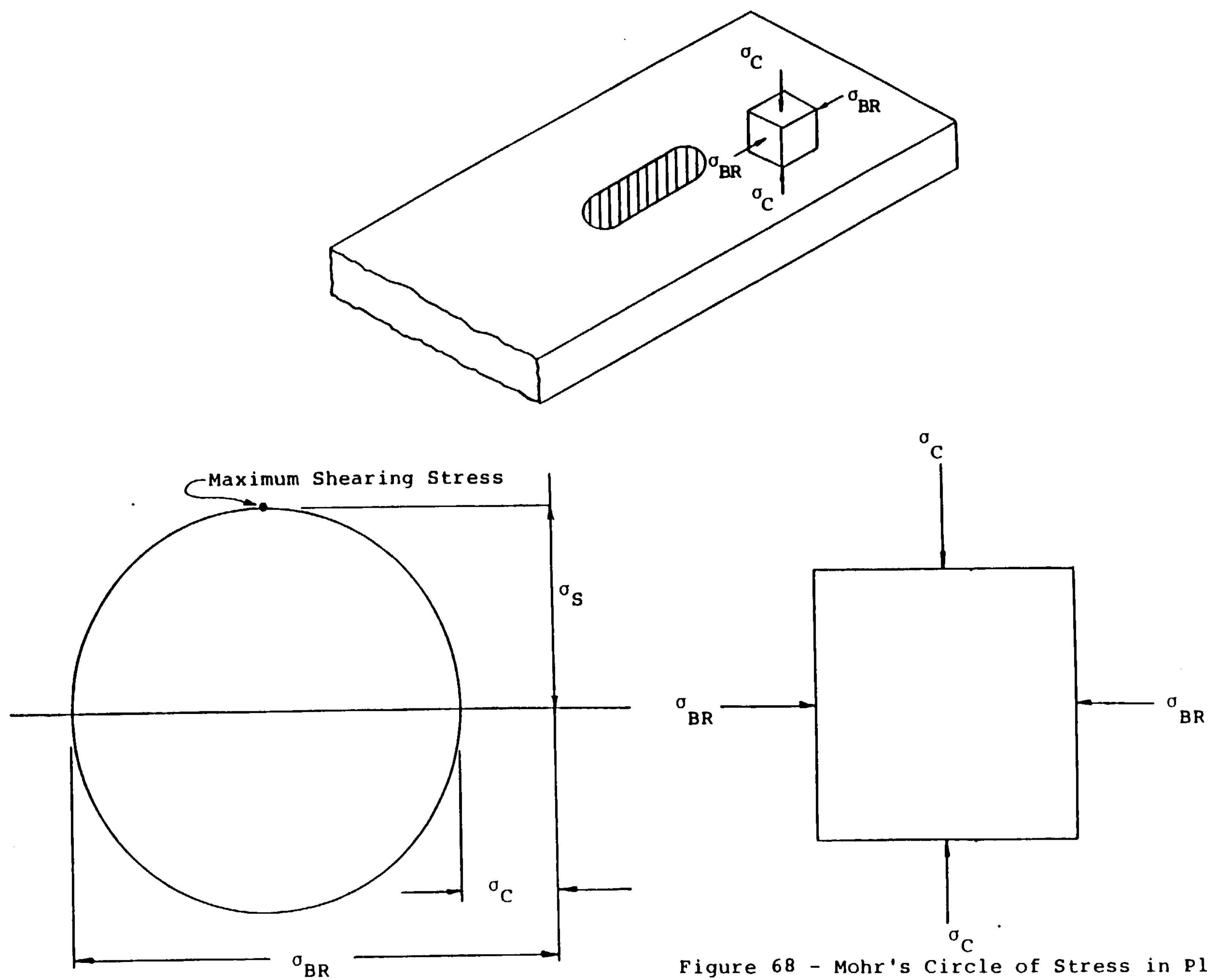


Figure 68 - Mohr's Circle of Stress in Plate

It should be stressed that a designer should not assume that the shear and bearing forces are divided uniformly across the appropriate part areas. Figures 2a, 2b and 2c have pointed out that the stress distribution varies widely across a bolt section.

From a design point of view, it is good to perform a good approximate stress analysis in order to avoid what could be a critical stress condition. Thus, if a calculation shows average shear stress to be critical, another bolt or two may be added. If a bolt bearing stress appears to be critical, the bearing area or the material bearing strengths may be increased or an insert used to increase the bearing area.

As shown in figure 69, bolted plates can fail not only in bearing but also in shear tearing if the bolt hole is close to the edge of the plate.

The shear tear out area (under a shear load V) is:

$$A = T_1 \cdot T_2 \cdot 2$$

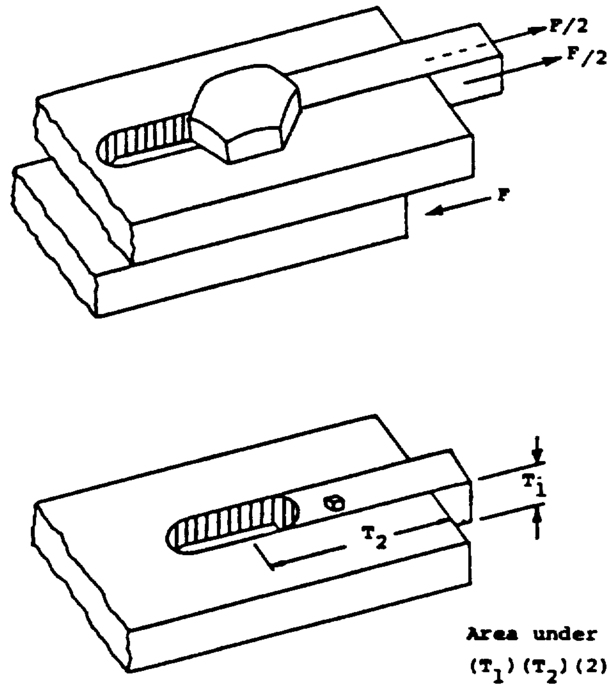
where T_1, T_2 are as shown in figure 69.

The shear stress is:

$$\tau_s = \frac{V}{A} k = \frac{V}{T_1 \cdot T_2 \cdot 2} \cdot k$$

where k is a shear factor (3/2 for a rectangular section). In most cases the compression preload stress is not added since it is usually small. However, if the compression prestress is critical, then it must be considered.

Holes or indents in plates are points of local stress concentration. An example of this is shown in figure 70. These stress concentrations could cause critical shear failure and should be avoided.



Area under shear =
$(T_1)(T_2)(2)$
$\sigma_S = F/A = F/(T_1)(T_2)(2)$

Figure 69 - Shear Tear Out

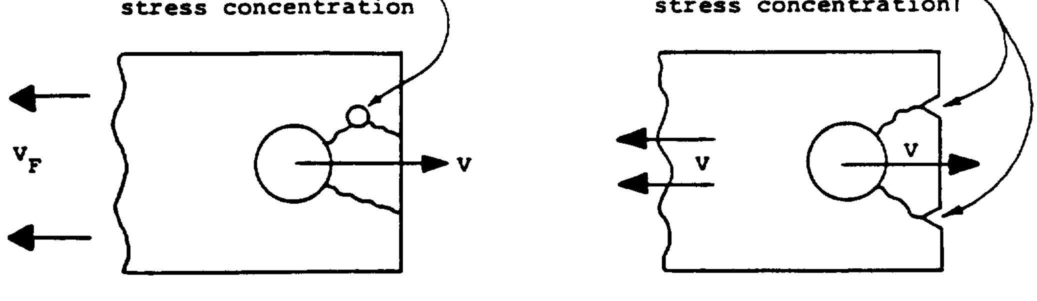


Figure 70 - Causes of Local Stress Concentration

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A third type of failure encountered in designs is where bolts tear plates in tension, as shown in figure 71. In addition, there are combined types of failure (for example, combined shear and tension, as shown in figure 72). Most failures of this type usually start as a material shear failure in which the shear stress is less than the tensile stress ($\sigma_S < \sigma_T$). As the load increases, the failure turns into a tear failure.

The shear tearout tensile stress is given by:

$$\sigma_{tension} = \frac{V}{A} = \frac{V}{2(T_1)(T_2)}$$

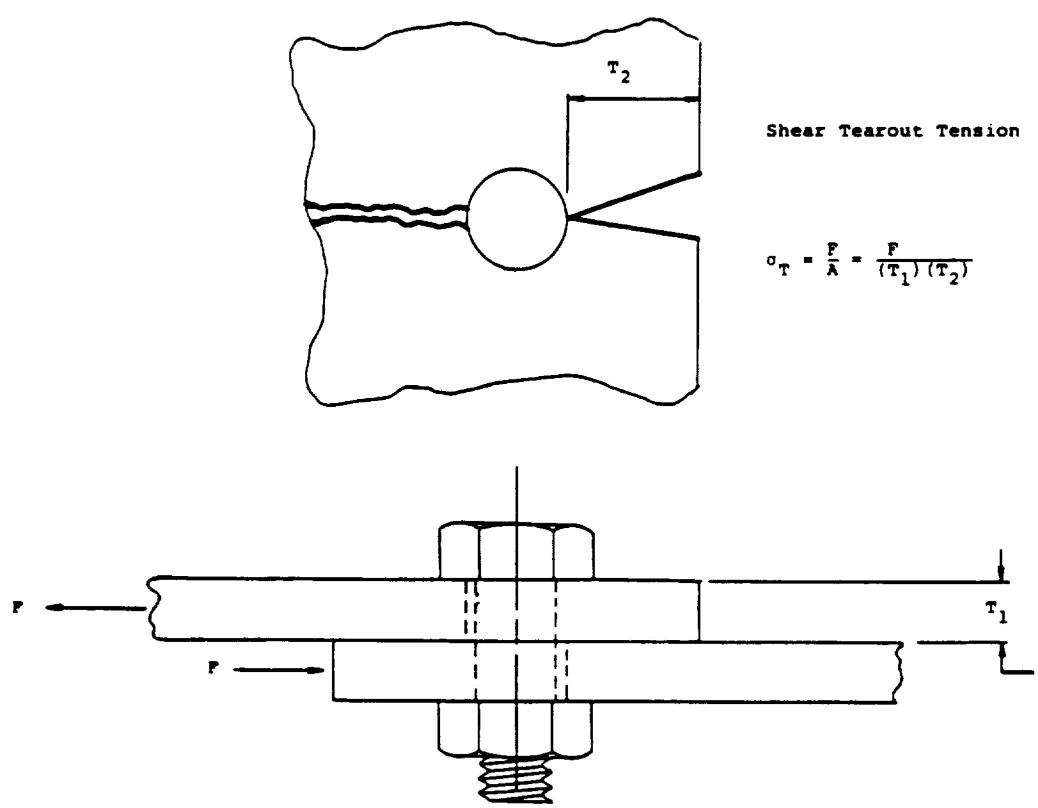

Figure 71 - Shear Tension tear-out

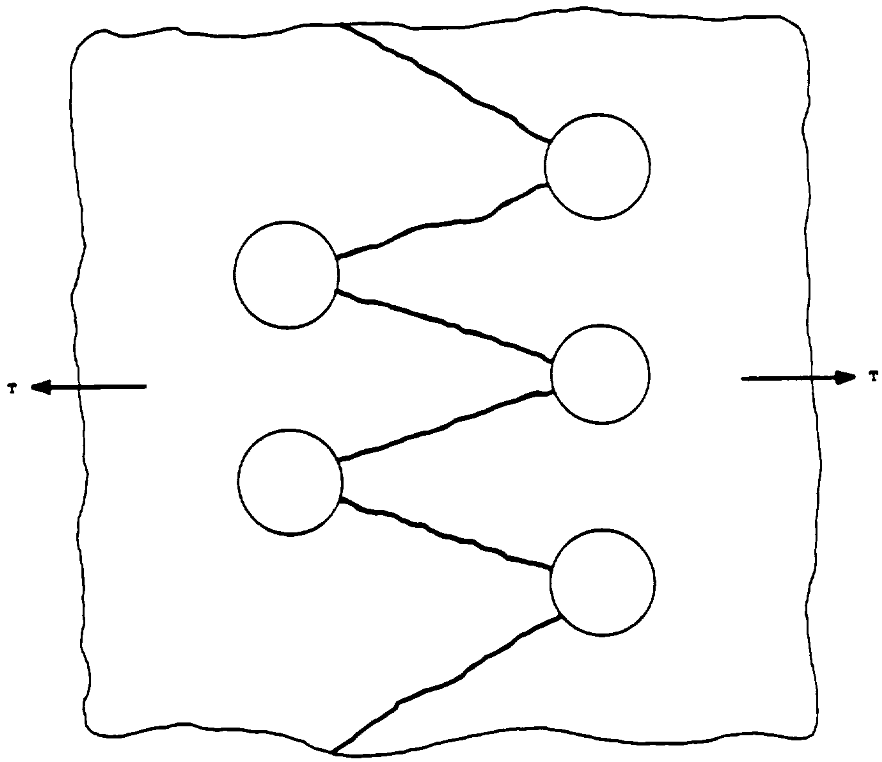

Figure 72 - Combined Shear and Tension

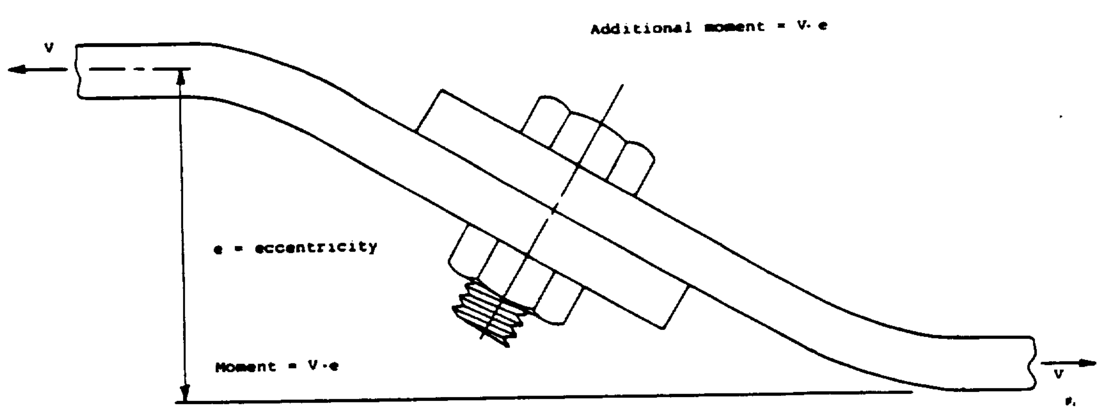

Figure 73 - Bending Shear Preload

All of the figures shown in this section have indicated a single lap joint under load. If the line of force V on one plate does not coincide with the line of force on the second plate (i.e., it is offset by an amount e), a moment is induced. (See figure 73.)

This type of offset load puts an additional tensile load on the bolted joint and on the bolt itself. This added tensile stress (due to flexural tension) combines with the shear stress due to the applied bolt shear load V_{bolt} and with the bearing load stress also due to the applied load V. The exact stress in this situation is difficult to predict.

A better joint (i.e., one which eliminates bending) is one which uses a double shear joint (see figure 74). This joint is easier to analyze and is therefore more predictable.

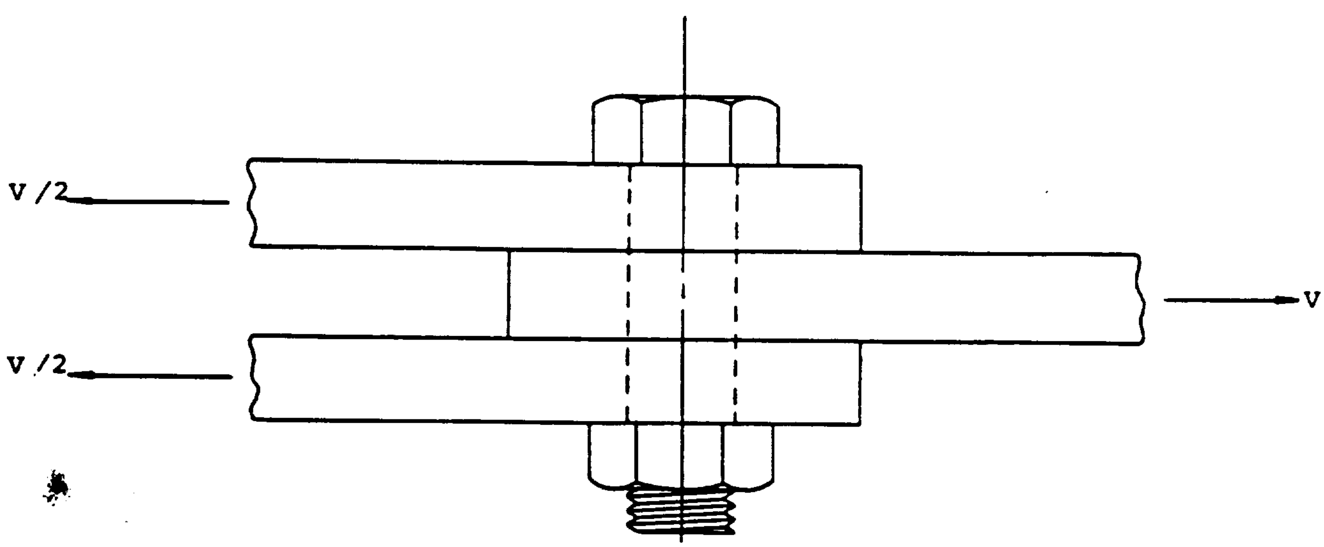

Figure 7 4 - A Properly Designed Joint

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SECTION VI

Locking Features in Bolts and Nuts

Q. 51 Why are locking features so important?

A. 51 It has been previously pointed out that one of the most serious problems in bolt and nut design is the gapping of plates or frames which are bolted together. This gapping and resulting impact of the two frames or plates together can cause shock loads that easily damage or destroy most materials (since most materials withstand only low shock stresses). Further gapping can even cause loosening of the nut and eventually loss of the nut with catastrophic consequences. Locking devices can prevent preload loss and nut loss during impact.

Q. 52 What principles should be followed in looking for a good technique for locking a nut on a bolt?

A. 52 First, the locking device should not introduce permanent damage to the bolt or its nut, nor to their threads. Secondly, the locking device should be reversible (i.e., removable). It was once said that letting bolts corrode into their nut was one of the best ways to lock the nut on. However, it is usually impossible to remove corroded nuts if removal is needed. Locking devices should be such that a standard torque method can be used to preload the bolt nut. Some companies list standard torques to be used with their locking nuts. If the locking device is damaged by a single use, it is important to mark or discard the nut so that it will not be re-used as a locking nut after its first torque-up.

Q. 53 What are some of the standard devices currently used for locking nuts?

A. 53 Any standard bolt or stud can be drilled through and then filled with a nylon plug or similar material plug which extends into the thread area. The plug bears into the nut thread areas and is then able to keep the nut from backing off. The nylon plug is never near the first several threads of the nut or bolt, and thus does not affect the critical stresses in these areas. (Figure 78)

There are also nuts which have a plastic ring molded to the top thread of the nut. This plastic is deformed as the nut is turned and the bolt threads penetrate the ring. This plastic deformation holds the nut in place.

Some companies fabricate a nut where the top one or two threads are deformed into a triangular shape. The nut literally springs outward on the bolt as it is torqued up and these last threads are placed in bearing. (Figure 79)

Some locking nuts have one thread in the nut made at a different angle than the other threads. The nut consequently locks as the nut is turned up. Again, it must be emphasized that this cannot be one of the critical threads in the nut or bolt. (Figure 80)

Q. 54 List some of the current technical literature available on locking devices.

A. 54 "How Locknuts Remove the 'Unknowns' in Joint Design," by Robert B. Aronson, Staff Editor, Machine Design, October 12, 1978, p. 164. [29]

"Prevailing Torque Locknuts--What they are and Where they Should be Used," by Robert A. Degen, Assembly Engineering, July 1979, p. 28, and June 1970, p. 30.[30]

[redacted] 95

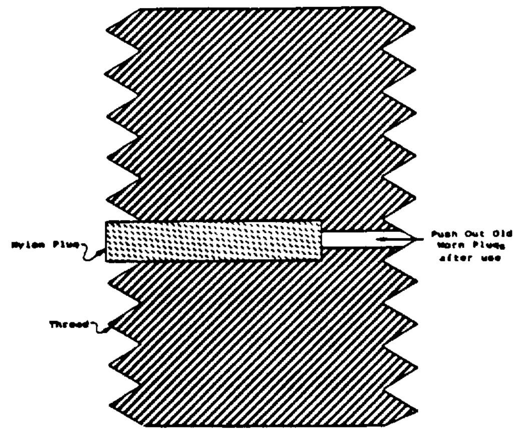


Figure 78 - Locking Device

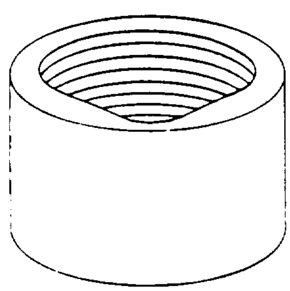


Figure 79 - Elliptical Thread

96 CD2

Manufacturing Problems

Q.55 What are some typical methods for manufacturing bolts?

A.55 Bolts are made by casting, machining, rolling, cold heading, forging, grinding, etc.

Q.56 What are some of the problems associated with these fabrication methods?

A.56 Books have been written on one or all of the problems associated with manufacturing of bolts and nuts. In this brief report, we can simply list several of the difficulties and recommend further references. Some of the problems include lack of proper quality control, improper steel batch controls, poor testing methods, poor thread fabrication methods, etc.

Q.57 What method of manufacturing is usually used for threads with critical bolted connections?

A.57 The rolling process. This is explained by a paragraph selected from Metals Handbook, Machining, prepared by ASM Handbook Committee, American Society of Metals, Metals Park, Ohio 44073:[31]

> "THREAD ROLLING (also known as roll threading) is a cold forming process for producing threads or other helical or annular forms by rolling the impression of hardened steel dies into the surface of a cylindrical or conical blank. In contrast to thread cutting and thread grinding, thread rolling does not remove metal from the work blank. Rather, thread rolling dies displace the surface metal of the blank to form the roots and crests of a thread.
>
> Dies for thread rolling may be either flat or cylindrical. Flat dies operate by a traversing motion. Methods that use cylindrical dies are classified as radial infeed, tangential feel, through feed, planetary, and internal. Each method is discussed in a separate section of this article..."

What are the principle advantages of rolled threads?

Any rolled, formed or forged type of operation forms the material without cutting through the grain structure. The grain structure is thus rolled to line up with the principle stresses. The critical stress region at the root of the thread is relieved by lining the grain of the metal up with the flow of the stress.

Q. 58 What other change in rolled threads has occurred in the past decade to affect the critical stress region?

A. 58 The radius at the root of the thread has been increased to increase the fatigue life and decrease the stress concentration. Thus the geometry has been changed and the methods of forming have been changed to help the designer.

Q. 59 What materials have been used and how has this helped?

A. 59 Many steels from 4130, 4340, 8630, 828, 303, 17-4 Ph, etc., are used for high strength bolts. Essential to the metallurgy and method of manufacturing is the necessity of having a large elongation after machining. Thus some of these materials end up with an elongation of 58% after machining and heat treatment. Others end up with an elongation of 12%. The higher elongations are necessary to eliminate problems of local bolt yielding.

Q. 60 What other critical problem comes from manufacturing?

A. 60 Hydrogen embrittlement due to improper coating, cleaning or plating of bolt surfaces. (See Hydrogen Embrittlement, Robert L. Spriat, and Dohn J. Laurilliard, Reprinted from Fasteners, Volume 21, No. 1, Industrial Fasteners Institute, Cleveland Ohio.)[32] An excerpt from this article indicates that:

> "...This phenomenon results from the absorption of atomic hydrogen-principally during electroplating--but also in acid pickling and alkaline cleaning operations prior to plating. In a process which is still not fully understood, it is thought that the atomic hydrogen is absorbed by and diffused through the metal, with a tendency to collect under load in areas of highest

stress concentration. There it can induce brittle failure, which in most cases is delayed and for which there is no means of advanced detection..."

With modern inspection techniques, along with better methods of manufacturing, hydrogen embrittlement is seldom a problem. It is good to keep up the guard, though, because without it methods of manufacturing could begin to degrade again. Be aware of older bolts manufactured during problem eras.

Q.61 Since hydrogen embrittlement is such a complex problem, list some references that can be used to get an elementary knowledge of the manufacturing problem.

A.61 How to Keep Hydrogen Embrittled Fasteners out of Assembled Products, Floyd S. Salser and David R. Kverek, Assembly Engineering, Sept. 1972.[33]

Why Titan's Bolts Failed, Louis Raymond and Ernest G. Kendall, Metal Progress, January, 1968.[34]

Hydrogen Embrittlement (see above reference).

The Right Fasteners are a Flyer's Best Friend, Edward F. Gowen, Sr., Approach, January, 1967.[35]

Radiused-Root Threads--Are they Really Better?, Robert L. Spriat, Richard A. Walker, Assembly Engineering, April, 1965.[36]

How Fasteners are Made, Thomas C. Baumgartner, Machine Design, January 9, 1969.[37]

Warning to the Users of Socket Screws, Precision Industrial Products Division, Standard Pressed Steel Co., Jenkintown, PA 19046.[38]

Mechanical Fasteners, Editorial, Materials Selector, Materials Engineering, Mid-October, 1967.[39]

Fatigue Cracks and What to do about Them, Clarence R. Smith, Assembly Engineering, August, 1972.[40]

Frontiers in Fastener Design, Clarence R. Smith, Assembly, Engineering, January, 1969.[41]

Metals Handbook, Machining, American Society for Metals, 8th Edition, Vol. 3, Metals Park, Ohio 44073.[31]

[redacted] 99

SECTION VII

Failure Analysis and Quality Control

Q.6 2 Why is quality control covered with failure analysis?

A.6 2 There have been and always will be failures in fasteners. All that a designer can do is to keep them to a minimum and be sure that the design inspection and quality control are comparable to the reliability needed for a particular job design. For example, if one bolt is used to suspend a man 100 feet in the air, the quality control is most important. If a thousand bolts are used to hold a machine together and one bolt failure would not stop or deter the operation of the machine quality control is not as important. What is most important is that the one fractured bolt be found in routine inspection and replaced.

Q.6 3 What are some of the methods used in quality control to prevent failure?

A.6 3 Some companies magnaflux (a patented electromagnetic inspection method) or x-ray all of their critical bolts. This is done routinely at many testing laboratories. Some manufacturers also coat their fasteners to prevent corrosion fatigue. Some manufacturers use special chemical treatments to prevent fatigue. Other manufacturers pull test bolt samples from vendor lots and inspect them from time to time to insure their quality. Most good bolt manufacturers remove samples from the vat of steel used to make a bolt lot and perform independent tensile specimen tests and chemical analysis on that vat batch.

Q.64 What are some of the non-destructive methods used to test bolts before sale?

A.64
- (a) Visual observation of discontinuities
- (b) Liquid dye penetrant inspection
- (c) Dry Magnaflux inspection (Magnetic particle)

100

(d) Electromagnetic testing

(e) Ultrasonic Inspection

(f) Lasar and other less used techniques

Reference:

Nondestructive Inspection and Quality Control, John E. Schleer, Metals Handbook, 8th Edition Vol. 11, American Society for Metals, Metals Park, Ohio, 44073.[42]

Q.65 What should the designer always keep in mind about quality control of bolt and nuts?

A.65 Quality control in a design will depend upon (1) the basic design, (2) the specifications for materials which are to be bolted as well as the fastener material, (3) the method of bolt manufacturing (including its quality control), (4) the method of inspecting, testing and quality control by the user for bolts he receives from his orders, (5) the amount of full testing of bolted assemblies where the load path is not well defined; testing where shock, vibration or other unknown loads affect the structure, (6) the experience and understanding of the mechanics who will assemble the structures, (7) the inspection and quality control of parts after assembly, (8) the level of field inspections during service including preventive maintenance inspection, and (9) methods of measuring loads and stresses during services to find fatigue problems.

Q.66 What difficult problem does the designer face in considering the above nine steps for quality control assurance?

A.66 A designer will not be able to control all of the work during the design. Further, he has to rely on others to carry out the job, and in many cases, the only control he will have is through specifications. He must rely on good mechanics to put the assemblies together. He has to make allowances when he knows that quality control will suffer.

[redacted] 14

The designer is constantly facing value engineering problems. He has to be able to justify his necessary reliability specifications, some of which may be quite expensive and challenged under a value engineering review. The clue to success in a design is to ask only for what is necessary. Take into account a balanced design--as quality control requirements drop off, less-expensive bolts could be used. If the weight or size is not critical, two or three inexpensive bolts are cheaper than one expensive bolt.

The designer must be aware of the manufacturer's abilities and note when his quality control is insufficient. A reduced quality control on the part of the manufacturer must be reflected in specifications and more stringent user quality control methods. The designer has to keep abreast of the latest in metallurgy technology. He must be aware of new inspection and quality control techniques for bolts and nuts.

Today, with so many quality control problems arising, it may be important to have methods for testing the final structure after assembly, prior to its use. In a joint part of a spacecraft there is only one final test before flight. In the case of automotive manufacturing, final inspection is a must with all manufactured parts.

The designer has to be aware of the limitations of everyone in the process chain. Remember that "a chain always fails at the weakest link."

Q.67 What are some references to these quality control and inspection problems?

A.67 Failures of Mechanical Fasteners, Walter L. Jenson, Metals Handbook, 8th Edition Vol. 10, American Society for Metals, Metals Park, Ohio 44073. (43)

Nondestructive Inspection and Quality Control. (See above)

In-Service Fastener Corrosion Tests, Robert T. Kelly, Editor, Assembly Engineering, October, 1968.(44)

Satellite Part Failure Laid to Complacency in Testing, Associated Press, January 20, 1979 (45)

Preventing Fatigue Failures, Clarence R. Smith, Assembly Engineering, April 1968. (46)

Fasteners that Fight Fatigue, Ronald Khol, Machine Design, February 20, 1975.(47)

What's Wrong with Fastener Specifications, Harry S. Grenner, Assembly Engineering, April 1969.(48)

Fastener Failures are Expensive, J. A. Trilling, Machine Design, December 14, 1972.(49)

Factors in Fastener Reliability, G. N. Hall, C. Eng. A.M.I. Mech. E., and Oliver Breward, Aircraft Engineering March 1967. (50)

Reflections, Technicalities and Smith-Kaboba, Clarence Smith, Assembly Engineering, May 1969. (also Jan. 1970) (51)

Fundamentals of High Strength Fasteners, Thomas C. Baumgartner, Standard Pressed Steel Co., Jenkintown, PA (52)

Failure Modes, Charles Lipson, Machine Design, Nov. 13, 1969, p. 222. (53)

Fatigue Design, Carl C. Osgood, Wiley-Interscience, John Wiley and Sons, Inc., New York, London and Toronto. (54)

Preventing Fatigue Failures, Clarence R. Smith, Assembly Engineering, March 1968.(55)

Design Considerations for Bolted Joints, John V. Liggett, Assembly Engineering, Decmeber 1968. (56)

Fastening Philosophy at GE...the Balanced Design Concept, Assembly Engineering, Nov. 1970. (57)

Threaded Fasteners, Fastening and Joining Reference Issue, Machine Design, Nov. 1975. (58)

High Grade Threaded Joints: Some Design Principles and New Developments, H. Ch. Klein, translated from German by Naval Ships Systems Command, Department of the Navy, October 1966.[59]

Some Problems of Fatigue of Bolts and Bolted Joints in Aircraft Applications, BUWEPS Report No. RAAE-343-62-1 Bureau of Naval Weapons, Washington, D.C.[60]

Standard Specification

Q.69 What is a standard specification for good commercial high strength allov steel bolts, nuts, threaded rod and washers?

A.69 Material: Alloy Steel

Tensile Strength: 160,000 psi up to and including screws of 5/16 in. dia.,
150,000 psi, up to 2 in. dia.

Elongation: 11% minimum

Hardness: Rockwell hardness C-34-40

Threads: All threads should incorporate increased root radii for improved fatigue life. All threads rolled.

Fillets: Increased fillets, under heads for improved head and fatigue strength.

Testing and Certification:

One bolt from every coil of wire shall be tensile tested according to Figure 5, or Figure 4, or Figure 3, or Figure 2, paragraph 4.4.3, page 148 of FF-S-86c. Elongation shall be measured with an extensometer or strain gage. Test results shall be submitted with the order. If the bolt does not meet the requirements listed above, three more bolts may be chosen at random and tested and an average of all three taken as the necessary requirement for testing according to this specification. The chemical analysis and official certification of the coil shall be included with the test results. The government reserves the right to randomly check the dimensions, hardness, elongation and strength of any bolt in the order. Failure of the screws (or bolts) to meet the requirement of this specification within 10% constitutes a reason for rejection of the entire order, within 90 days of receipt of order. In this specification the words screw and bolt shall mean the same thing and they can be used interchangeably.

Finished Hexagon Nuts

Specifications:

Tensile Strength: Maximum 130,000 psi. Minimum 110,000 psi.

Hardness: RB 88-102

Threads: Class 2. All threads incorporate increased root radii for improved life.

Nut Length: The nut length shall be a minimum equal to the diameter of the bolt. (Thus in a 1/4 in colt, the length of the nut shall be 1/4 in (min.)

Performance: The nut shall be required to meet the full strength of the cap screw, based on multiplying the full strength of the cap screw, based on multiplying the nominal minimum area of the bolt by 160,000 psi for cap screws 5/16 and under and by multiplying the nominal minimum area of the bolt by 150,000 psi for cap screws above 5/16 inches in diameter.

Testing: The government reserves the right to randomly check the dimensions, hardness and strength of any nut to meet the requirements within 10%. Failure to meet these requirements within 10% constitutes a reason for rejection of the entire order, within 90 days of receipt of order.

Threaded Rod

Specification:

Material: Alloy Steel

Tensile Strength: 135,000 psi within 15%

Hardness: Rc 32-38

Elongation: 12%

Threads: Class 2

Finish: Depending on use

Testing: See testing under Cap Screws except for the requirements of 135,000 psi tensile strength and RC 32-38, elongation 12%

Washers

Tensile Strength: 160,000 psi to 220,000 psi

Hardness: Rockwell c 40 to c 45

Thickness: Washer thickness/Cap Screw Diameter is a minimum of .160 ratio

Diameter: The outside diameter of the washer/Cap Screw diameter is a minimum of 1.65

Summary:

Many of the major problems encountered in bolted joint designs have been explored in this report. With careful re-examination of these problems and their recommended solutions, the designer can produce bolted joint design of high performance and structural integrity. As an aid to these recommendations, a summary of the questions posed in this report are provided at the beginning of this report for review and a quick check list. This list should be reviewed for each project design.

MEMO: 22 SEPT. 1981

TO: R. McDONNELL - CODE 741

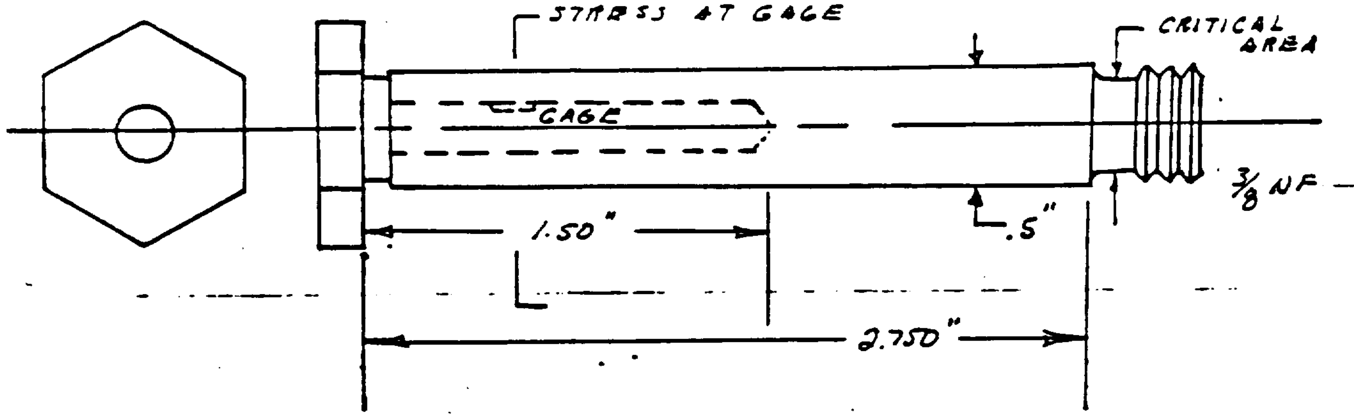
FROM: J. KERLEY CODE 754.1

SUBJECT: BOLTS TESTED AND USED IN G.A.S. EXPERIMENTS.

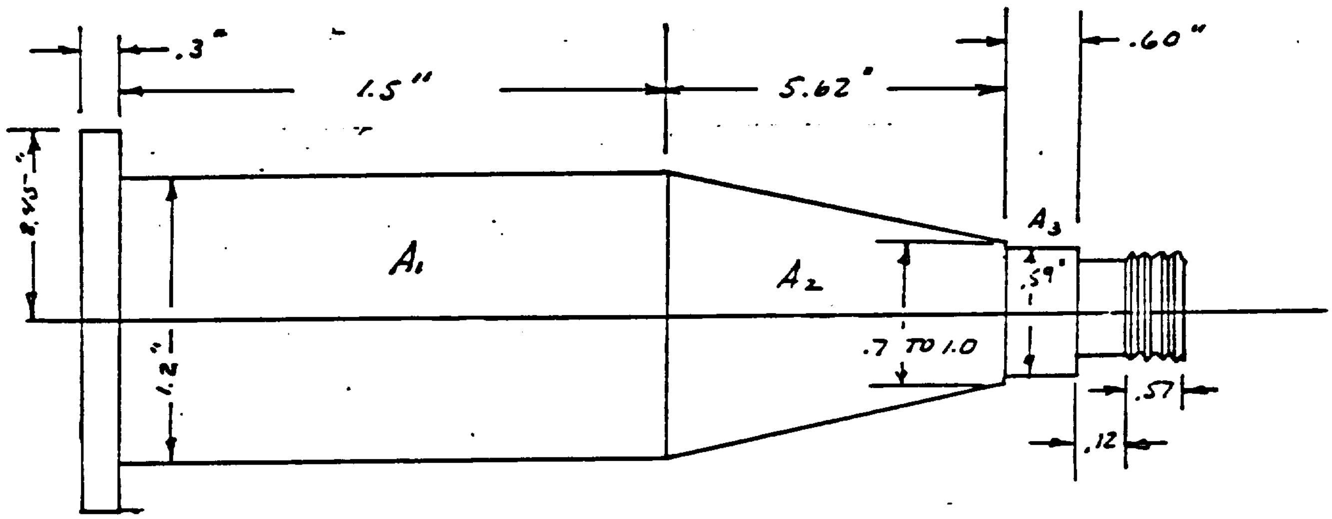
SUMMARY:

TESTING AND ANALYSIS OF ADAPTER BEAM/ ORBITER TIE DOWN BOLTS USED IN G.A.S. DEMONSTRATE HIGH STRESSES UNDER FLIGHT LOADING CONDITIONS. THESE LOADS ARE CAUSED BY A COMBINATION OF PRE-STRESS AND DYNAMIC LOADING DURING LAUNCH. SOME STRESSES PUT THE BOLTS IN THE YIELD REGION AND OTHER STRESSES DEMONSTRATE A POSSIBILITY OF FAILURE. TO VERIFY THIS DATA, MODELS OF THE ACTUAL BOLTS WERE FABRICATED - STRAIN GAGES INSTALLED IN THE BOLTS - TORQUED - AND VIBRATED ON THE GSFC C-220 SHAKER SIMULATING THE ACTUAL FLIGHT LOADING CONDITIONS INCLUDING THE STRESSED DUE TO GRAVITY AND BLAST-OFF. THE RESULTS OF THIS ANALYSIS SHOW A POSSIBILITY OF STRESSES AS HIGH AS 444,100 P.S.I. FOR G.A.S. 1000 lb PAYLOADS. $M.S. = \frac{180{,}000}{444{,}100} = -.405$

TEST RESULTS AND SUBSEQUENT ANALYSES FOLLOWS:

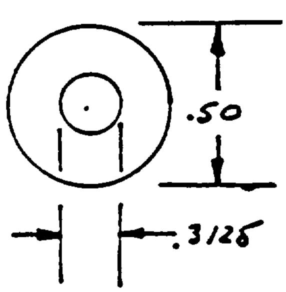


TEST BOLT - HEAT TREAT TO 150,000 #/in
ELONGATION = 18% - 4340



TYPICAL ACTUAL BOLT

TEST BOLT - DEFLECTION + K_b WITH 10,000#



$A_1 = .1964 - .0767 = \underline{.120}$ □"

$A_2 = \underline{.196}$ □"

$$\delta = \frac{PL_1}{A_1E} = \frac{(10{,}000)(1.5)}{(.120)(30\times10^6)} = .004$$

$$+ \frac{PL_2}{A_2E} = \frac{(10{,}000)(1.250)}{(.196)(30\times10^6)} = .002$$

$$\Sigma = .006''/10{,}000\#$$

$$K_b = \frac{EA}{L} = \frac{FORCE}{\Delta L} = \frac{10{,}000}{.006} = \underline{\underline{1.67\times10^6\ \#/in}}$$

DEFLECTION OF FLIGHT BOLT

$$A_1 = \frac{(1.2)^2\pi}{4} = 1.131^{\square''}$$

$$A_2 = .950^{\square''}$$

$$A_3 = .273^{\square''}$$

$$\delta = \frac{PL}{AE} = \frac{(10{,}000)(1.5)}{(1.13)(30\times10^6)} + \frac{(10{,}000)(5.62)}{(.95)(30\times10^6)}$$

$$+ \frac{(10{,}000)(.6)}{(.273)(30\times10^6)} = .00044 + .00198 + .00073$$

$$= \underline{\underline{.00315''}}$$

$$K_b = \frac{EA}{L} = \frac{\text{FORCE}}{\Delta L} = \frac{10{,}000}{.00315} = \underline{\underline{3.17\times10^6}}\ \#/\text{in}$$

DEFLECTION OF TEST BOLT 2 TIMES FLIGHT BOLT. COMPATIBLE FOR TESTING.

Two different (1/2 x 3/8 -24) shoulder bolts WERE PRE-TORQUED, while strains WERE MEASURED:

The calculations AND Test Results of the bolts WERE checked. ONE checked in AT 853 $\mu\epsilon$ WITH A CALCULATED STRAIN OF 947 $\mu\epsilon$ (10%). The other checked in at 1012 $\mu\epsilon$ WITH A CALCULATED STRAIN

[redacted]

of 947 $\mu\epsilon$ (57%) A SUMMARY OF DATA FOLLOWS - DEMONSTRATING THE FACT THAT PROPER PRE-TORQUING IS A GOOD MEASURE OF CLAMPING FORCE.

STRESSES USED TO check TORQUING:
20,000 #/□" - 30,000 #/□" - 40,000 #/□"

μ = (COEFFICIENT OF FRICTION) = .15
THREADS LUBRICATED WITH MOLLY/DISULPHIDE

20,000 #/□" WITH a μ of .15 is 116 "# TORQUE
20,000 #/□" ON A 3/8-24 thread is 1,700 #

30,000 #/□" μ = .15 is A TORQUE OF 173 "#
30,000 #/□" on A 3/8-24 thread is 2,550 #

40,000 #/□" μ = .15 is A TORQUE OF 240 "#
40,000 #/□" ON A 3/8-24 thread is 3,400 #

AREA AT STRAIN GAGE = .120 □" (PAGE 2)

20,000 #/□" (stress at threads) $\frac{1700\#}{.1196}$ = 14,200 #/□" (STRESS AT strain gage and not in threads)

30,000 #/□" (stress at threads) $\frac{2550\#}{.1196}$ = 21,300 #/□" (STRESS AT GAGE)

40,000 #/□" (stress AT THREADS) $\frac{3,400}{.1196}$ = 28,400 #/□" (STRESS AT GAGE) [redacted]2

$\epsilon = \frac{\sigma}{E}$ CALCULATED STRAINS.

$$\epsilon_{20,000} = \frac{14,200}{30 \times 10^6} = .00047 = \boxed{470\ \mu\epsilon}$$

$$\epsilon_{20,000} = \frac{21,300}{30 \times 10^6} = .00071 = \boxed{710\ \mu\epsilon}$$

$$\epsilon_{40,000} = \frac{28,400}{30 \times 10^6} = .000947 = \boxed{947\ \mu\epsilon}$$

CALCULATED STRESS IN THREADS	CALCULATED STRESS IN SHANK	CALCULATED STRAIN IN SHANK	TESTED STRAIN IN BOLT #1	TESTED STRAIN IN BOLT #2
20,000 #/□"	14,200 #/□"	470 με	368 με (22%)	446 με (5%)
30,000 #/□"	21,300 #/□"	710 με	595 με (16%)	737 με (4%)
40,000 #/□"	28,400 #/□"	947 με	853 με (10%)	1012 με (7%)

The CALIBRATED bolts WERE calibrated ON A TINEOUS Olsen 60,000# TENSION TESTING MACHINE. THE GAGES IN THE bolts CHECKED OUT TO ± 5%. The TINEOUS OLSEN testing MACHINE HAS been checked WITHIN THE LAST YEAR by GAGES TRACEABLE TO THE BUREAU OF STANDARDS.

 [redacted]

THE CRITICAL FLIGHT BOLT DIAMETER FROM DRAWING NO. V073-340120 — ROCKWELL INTERNATIONAL CORPORATION 1/13/77 IS: $.311'' - .010'' = .301''$

AREA = $.071^{\square''}$

MAX TORQUE = $400''^{\#}$ (LATER CHANGED TO $420''^{\#}$)

WITH A $\mu = .15$

LOAD ON 3/8-24 (NF) THREAD = $5,800^{\#}$

(REF. ANALYSIS OF NUT + BOLT TORQUES
JAMES E. FOISY
G.E. REPORT R64SE45 JULY 1964)

STRESS DUE TO PRE-TORQUE = P/A

$$\frac{5,800}{.071} = 81,690\ ^{\#}/_{\square''}$$

<u>A FLIGHT BOLT WITH A μ OF .15 CAN BE TORQUED UP TO $81,690\ ^{\#}/_{\square''}$ BEFOR FLIGHT.</u>

$$K_{bolt} = \frac{EA}{L} = \frac{FORCE}{\Delta L}$$

$$K_{TEST\ BOLT} = \frac{10,000}{.006} = 1,666,000\ ^{\#}/_{in} = (1.67 \times 10^6\ ^{\#}/_{in})$$

$$K_{FLIGHT\ BOLT} = \frac{10,000}{.00315} = 3,170,000\ ^{\#}/_{in} = (3.17 \times 10^6\ ^{\#}/_{in})$$

THIS IS A GOOD SIMILITUDE FOR TESTING AS THE FRAME IS MUCH MORE PLIANT.

STRAIN IN BOLT DURING TEST OF SIMULATED LAUNCH = 1103 $\mu\epsilon$ IN FWD. BOLT. STRESS = $33,000\ ^{\#}/_{\square''}$ AT GAGE IN FWD. BOLT.

LOAD = $(33,000)(.12) = 3,960^{\#}$ (FWD BOLT.) $(\sigma_{AT\ GAGE} \times A.)$

<u>STRESS IN CRITICAL SECTION:</u>

$$\frac{3,960}{.071} = \underline{\underline{55,775}}\ ^{\#}/_{\square''} = \frac{\text{LOAD IN BOLT}}{\text{MIN. AREA AT CRITICAL SECTION.}}$$

THIS STRESS IN BOLT IS NOT ADDED DIRECTLY TO STRESS DUE TO PRE-TORQUING. HOWEVER IN SHUTTLE CASE THE FRAMES ARE LESS STIFF THAN THE BOLT. THE BOLT, THEN, WILL TAKE MOST OF THE $55,775\ ^{\#}/_{\square''}$ ON TOP OF THE PRE-LOAD

BOLT NO.	LOCATION	GAP* (in)	TORQUE (in-lb)
-002	FWD.	.024	240
-006	SILL	.021	450
-009	AFT	.024	210

STIFFNESS OF BOLTS & TEST FIXTURE DURING TEST.

$K_{b\,(TEST)} = \frac{EA}{L} = \frac{FORCE}{\Delta L} = \frac{10,000}{.006} = 1.67 \times 10^6$ #/in

$K_{FRAME\,(ALUM)} = \frac{10,000}{\Delta L}$

HOLLOW CENTER

$\frac{1}{2}$" DIA = .2 □"

2" DIA = 3.14 □"

1" DIA = .785 □"

AV. ON TOP:

$$\frac{\begin{matrix}.2 \\ 3.14\end{matrix}}{3.34/2} = 1.67 \;\square''$$

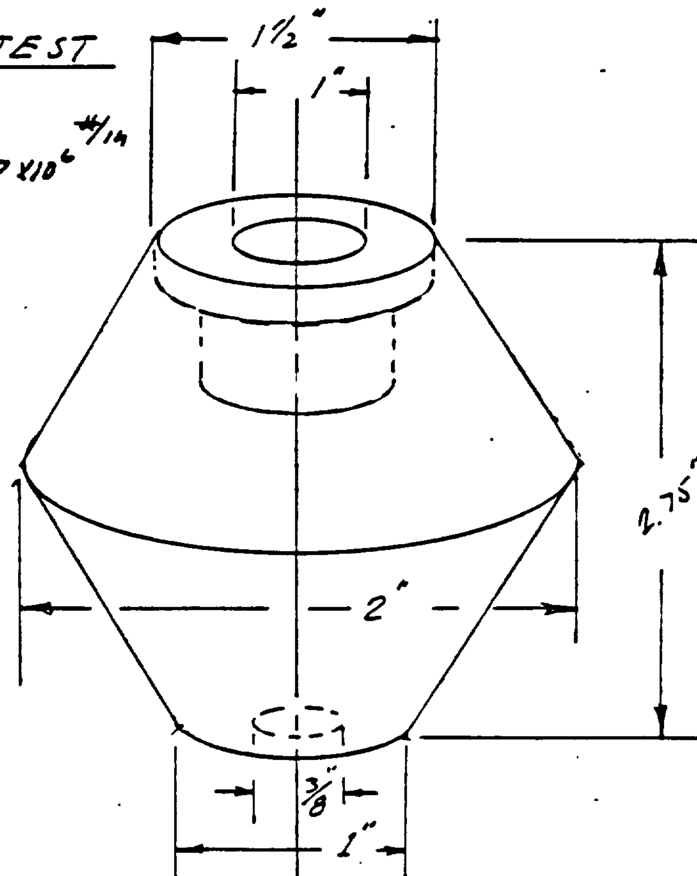
AV. ON BOTTOM:

$$\frac{\begin{matrix}.785 \\ 3.14\end{matrix}}{3.92/2} = 1.962 \;\square''$$

$$\delta = \frac{PL}{AE} = \frac{(10,000)(1.375)}{(1.67)(10 \times 10^6)} + \frac{(10,000)(1.375)}{(1.962)(10 \times 10^6)} =$$

$.00082 + .0007 =$

$.0015$ in.



CALCULATION OF EFFECTIVE AREA

SEE FIG. 27-B - PAGE 3-20 NUTS & BOLTS by J. KERLEY NASA REPORT - JAN 1980

$K_{FRAME} = \frac{10,000}{.0015} = 6.67 \times 10^6$ #/in

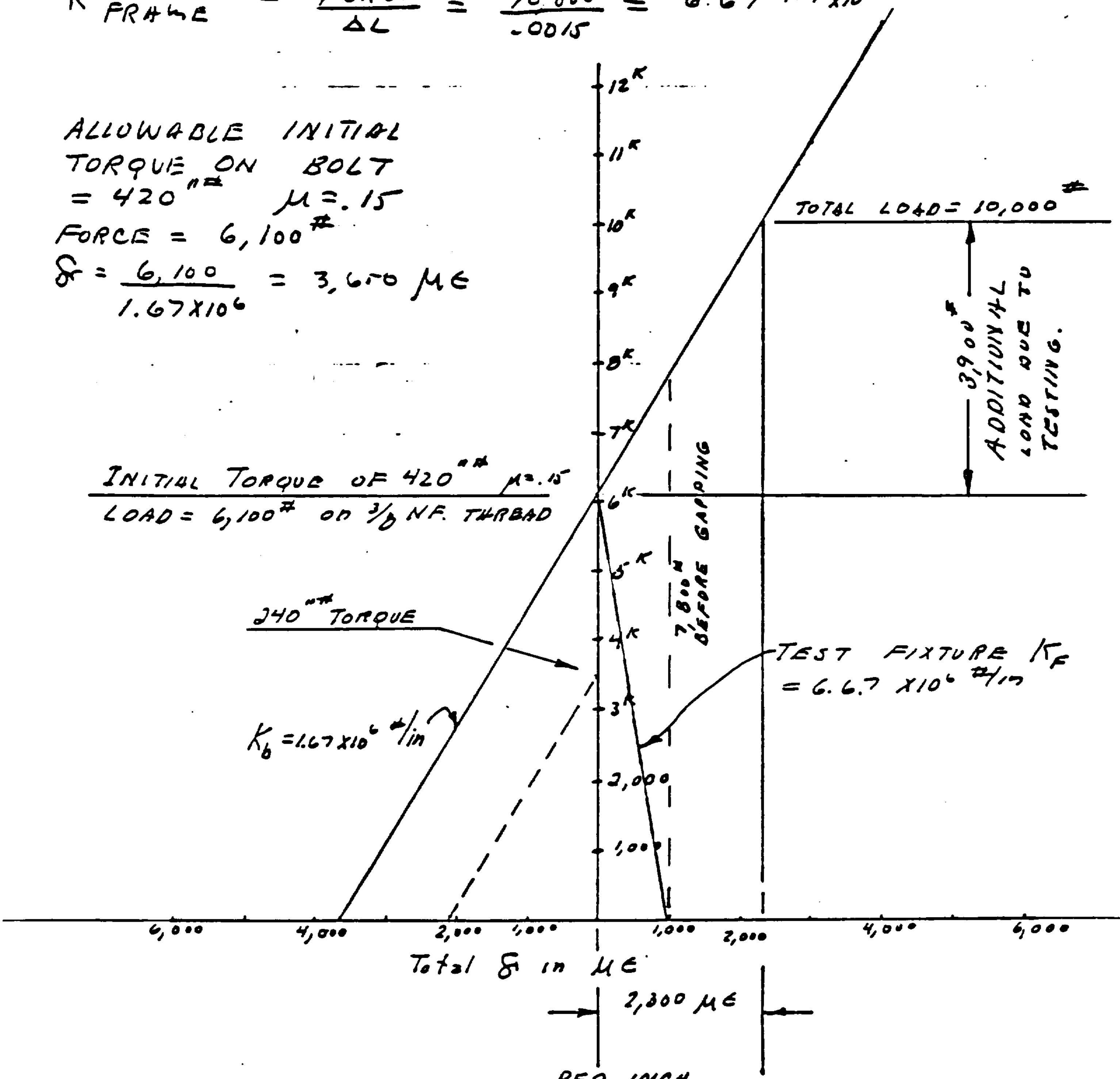
STIFFNESS IN BOLT FOR TEST $= K_b = 1.67 \times 10^6$ #/in
IT TAKES 1,670,000# TO MOVE BOLT 1.0"

δ FRAME OVER 2.75" = .0015 in.

$$K_{FRAME} = \frac{FORCE}{\Delta L} = \frac{10{,}000}{.0015} = 6.67 \text{ \#/in} \times 10^6$$

ALLOWABLE INITIAL TORQUE ON BOLT
= 420 in# $\mu = .15$
FORCE = 6,100#

$$\delta = \frac{6{,}100}{1.67 \times 10^6} = 3{,}650\ \mu\epsilon$$



RECORDED $\mu\epsilon$ in bolt = 1,100 $\mu\epsilon$ Locally. Total DEFLECTION in bolt = 1,100 $\mu\epsilon$ X 2.09" (EFFECTIVE LENGTH)* = 2,300 $\mu\epsilon$ TOTAL DURING INITIAL TEST GAPPING PROBABLY OCCURED IN TEST FIXTURE AND THE FRAME LOST ITS PRESTRESS OF 6,100# MOMENTARILY.

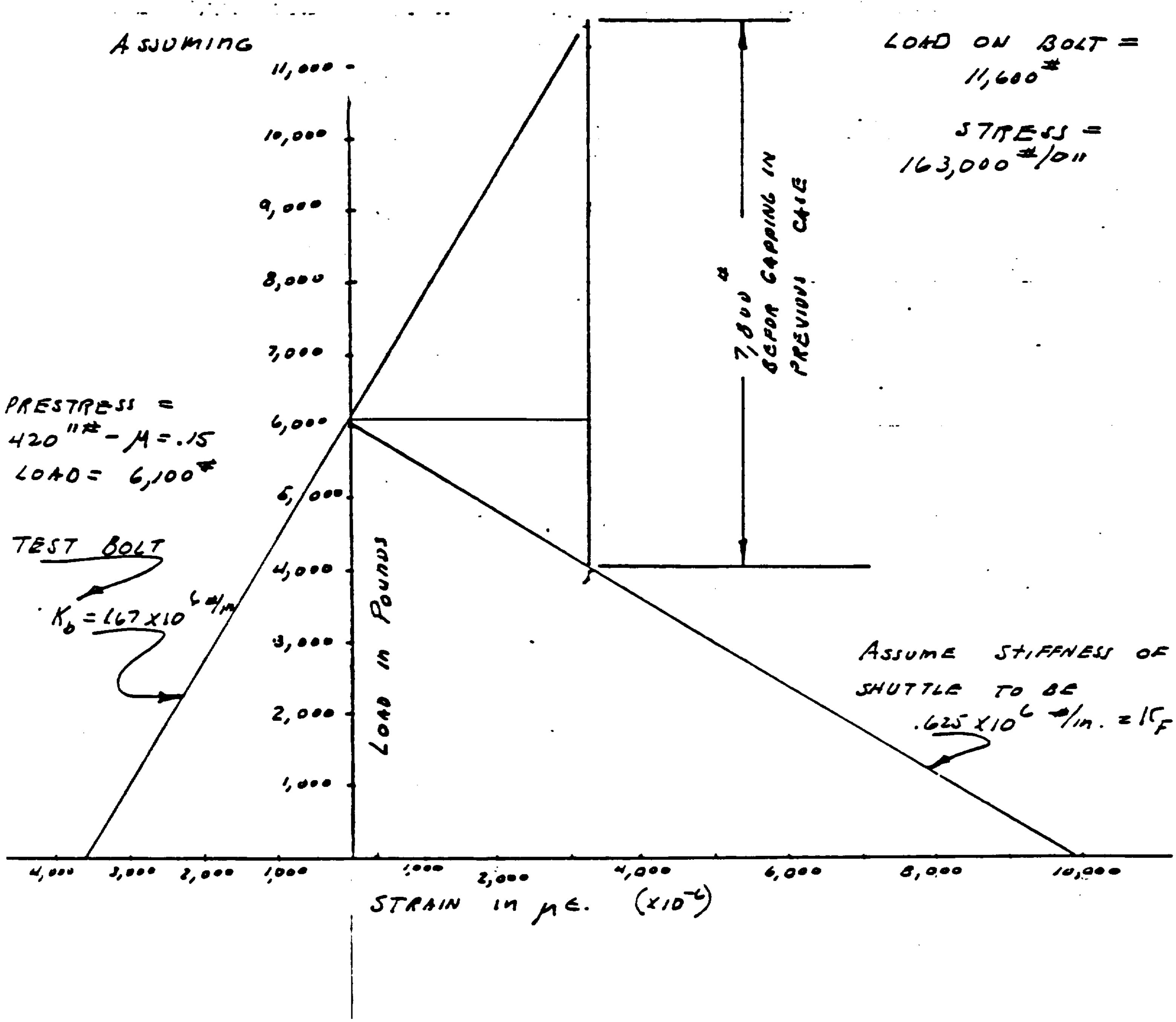
* STRESS DISTRIBUTION VARIES DOWN THE BOLT BECAUSE OF CHANGES IN THE CROSS-SECTION.

 [redacted]

IN ACTUAL SHUTTLE LOADS THE FRAME WILL LOSE MUCH OF ITS STIFFNESS. IT IS not KNOWN how much AS IT WAS NEVER MEASURED.

CALCULATED STIFFNESS OF TEST BOLT IS:

$\frac{10{,}000^{\#}}{.006} = 1.67 \times 10^{6}\ {}^{\#}/in$ 1.67 million pounds moves THE BOLT 1.0"



TEST BOLT SIMULATED IN SHUTTLE

 [redacted]

ASSUME BOLT LOAD ≅ MOMENT

ANALYSIS OF ADDITIONAL LOADS ON BOLTS FOR G.A.S. PAYLOADS

1. MULTIPLY LOADS BY $2\frac{1}{2}$ (1000# VS. 400#)
2. C.G. MOVED FROM 17" TO 21" = 1.2 TIMES THE MOMENT
3. ASSUME ORIGINAL LOAD = 3,900#
4. NEW LOAD = (3,900)(2.5)(1.2) = 11,700#
5. SHUTTLE BOLT HAS $K_b = 3.24 \times 10^6$ #/in.
6. ASSUME INITIAL TORQUE TO BE 420"# & $\mu = .15$
7. INITIAL PRE-TORQUE GIVES A LOAD OF 6,100#
8. ASSUME SHUTTLE STIFFNESS = $.625 \times 10^6$ #/in.

NOTE: AFTER PRE-STRESS of 6,100# MOST OF THE ADDITIONAL LOADS GO TO THE BOLT, AND NOT THE FRAME. 1,900# GOES TO THE FRAME AND 9,800# GOES TO THE BOLT.

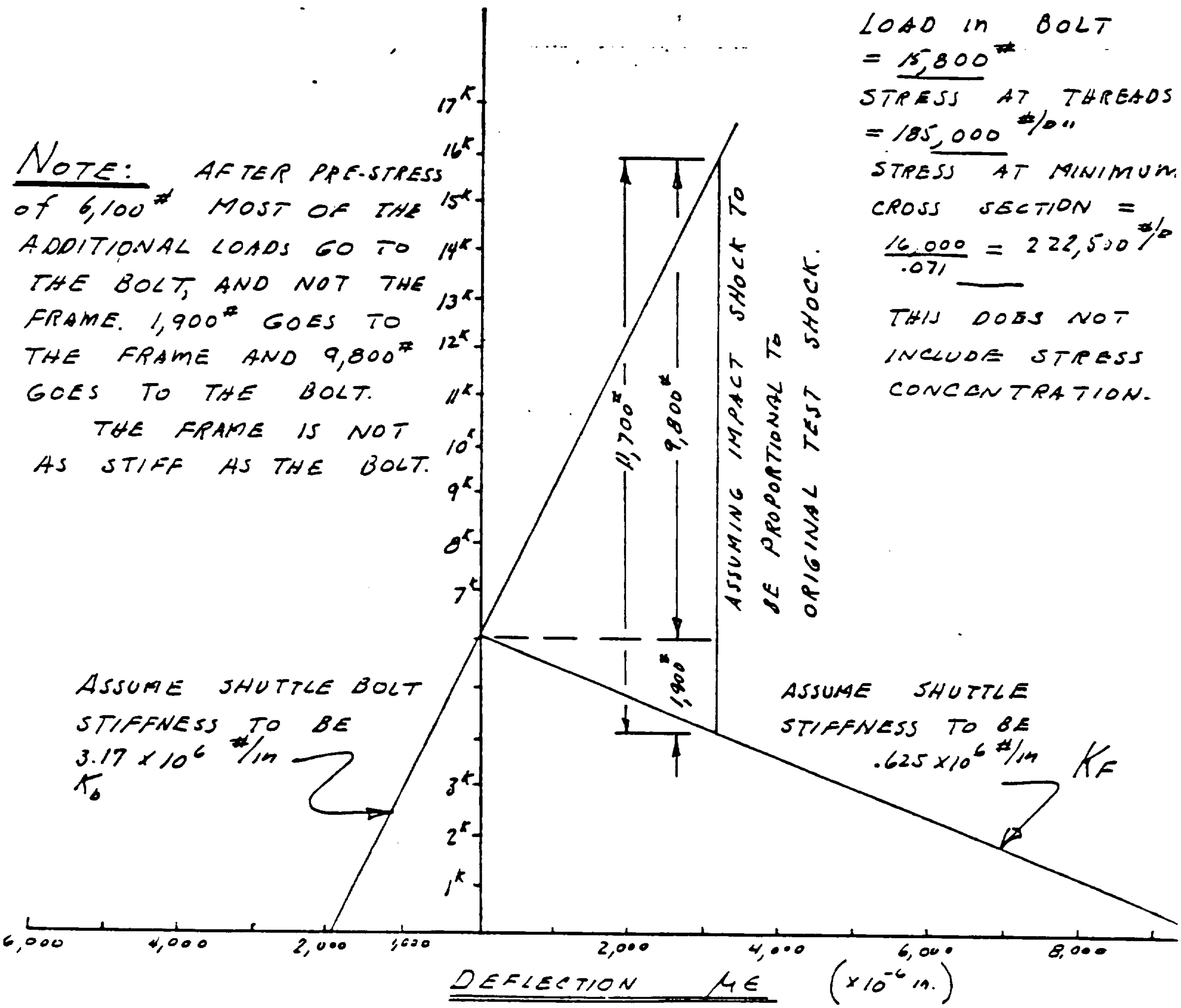
THE FRAME IS NOT AS STIFF AS THE BOLT.

LOAD IN BOLT = 15,800#

STRESS AT THREADS = 185,000 #/□"

STRESS AT MINIMUM CROSS SECTION = $\frac{16,000}{.071}$ = 222,500 #/□"

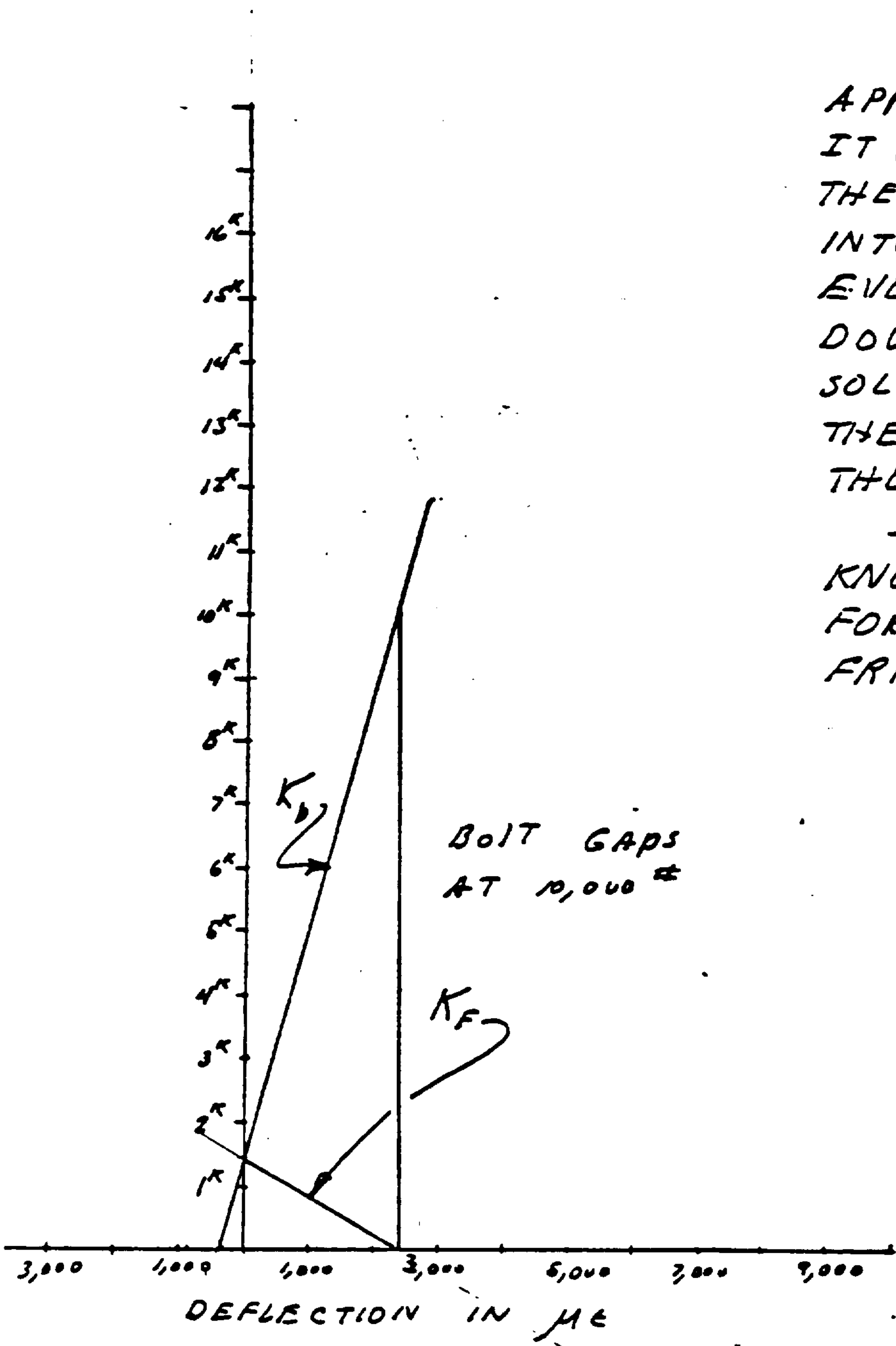
THIS DOES NOT INCLUDE STRESS CONCENTRATION.



DEFLECTION μE ($\times 10^{-6}$ in.)

 [redacted]

THE FOLLOWING ANALYSIS IS FOR AN ACTUAL BOLT ON THE SHUTTLE - PRESTRESSED TO 100"# $\mu = .15$
PRE-LOAD = 1,400#



APPLIED LOAD = 11,700#
IT GAPS AT 10,000# + ALL OF THE ADDITIONAL LOAD GOES INTO THE BOLT.
EVEN BRINGING THE TORQUE DOWN TO 100"# DOES NOT SOLVE THE PROBLEM. WHEN THE BOLT GAPS IT TAKES THE ENTIRE LOAD.
IT IS DIFFICULT TO KNOW WHAT THE CLAMPING FORCE IS FOR LOW TORQUES FRICTION IS TOO PROMINENT

CIRCUMSTANCES THAT COULD CHANGE VALUES.

1. THE VALUE OF μ COULD BE HIGHER THAN .15 AND THE PRE-STRESS WOULD GO DOWN. IT HAS BEEN SHOWN THAT THIS DOES NOT HELP MUCH

2. WHILE THE TORQUE-LOADING IN THE TEST BOLT WAS MEASURED - IT WAS NOT MEASURED IN THE FLIGHT BOLT.
   THIS SHOULD BE MEASURED AND IT COULD BE DONE WITH LESS EXPENSIVE MATERIALS SUCH AS STEEL + THE VALUES FOR 718 INCONEL CALCULATED.

3. THE STIFFNESS OF THE SHUTTLE COULD CHANGE. THIS SHOULD BE MEASURED OR CALCULATED FROM EXISTING DWGS.

4. THE LOADING IN THE SHUTTLE MAY BE DIFFERENT THAN THAT SEEN IN THE TEST RESULTS, BUT THERE IS NOTHING IN THE ANALYSIS OR TESTING TO INDICATE THAT IT MAY BE LESS.

5. THE LOAD BEING A SHOCK PULSE MAY CAUSE SLIGHT YIELDING + SIMPLY LOOSEN THE PRE-TORQUE. THIS COULD BE MORE DAMAGING TO STRESS CONCENTRATION POINTS, SUCH AS THE RADIUS ABOVE THE THREADS. IT SHOULD BE TESTED.

6. THE LOCKING FEATURE OF THE NUT PLATE WILL CAUSE A REDUCTION IN EFFECTIVE TORQUE AND A REDUCTIDIN IN CLAMPING FORCE. MOST SIMILAR DEVICES CAUSE A (10-15)% REDUCTION IN HIGH TORQUES.

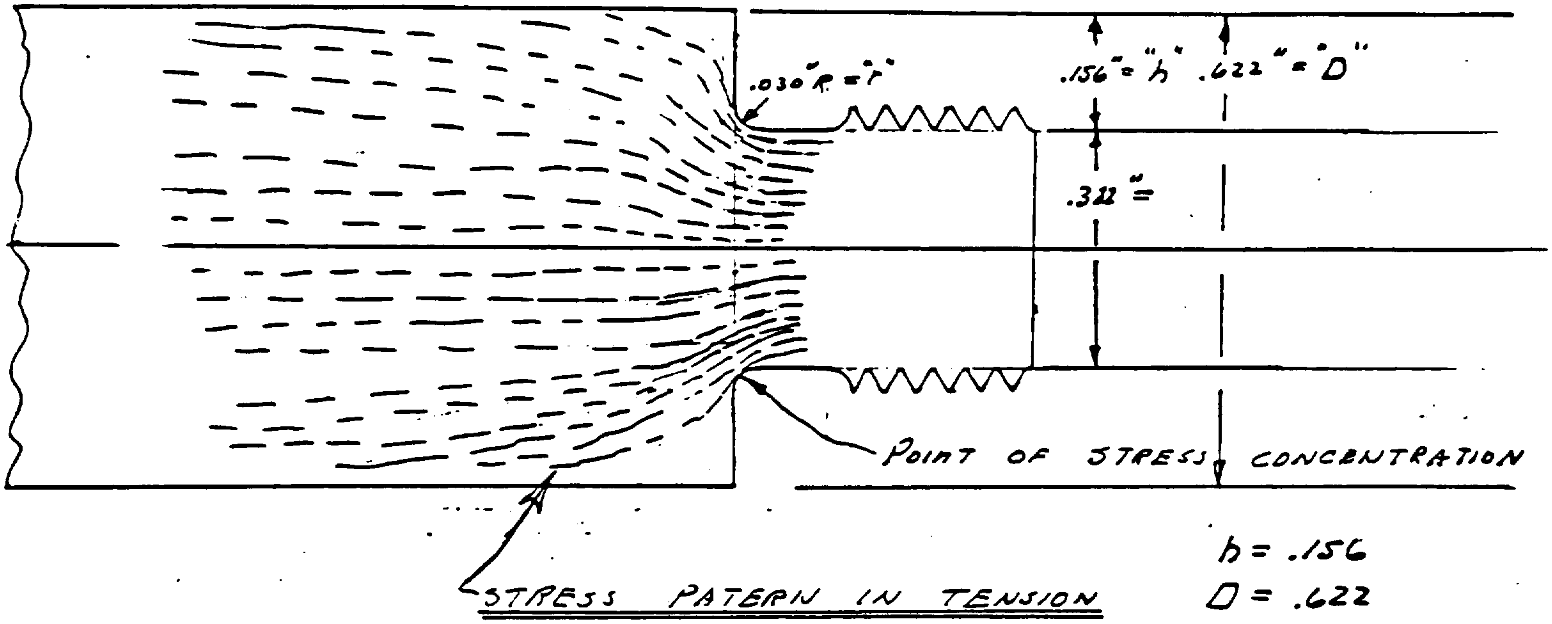
7. FLIGHT CONDITIONS IN THE NEXT ORBITER FLIGHT COULD BE WORSE. LANDING <u>COULD</u> BE A CRITICAL LOAD CASE.

 [redacted]

STRESS CONCENTRATION AT TRANSITION OF BOLTS:

SCALE: 4 TO 1 (TO SCALE)

-.009 PART DWG. NO. V073-340120
(ROCKWELL INTERNATIONAL CORP.)



$h = .156$
$D = .622$
$r = .030$

REFERENCES:

① PETERSON, R.E.; - STRESS CONCENTRATION FACTORS, John WILEY + Sons Inc., 1974

② FESSLER, H., C.C. ROGERS, + P. STANLEY: SHOULDERED PLATES AND SHAFTS IN TENSION AND TORSION, J. STRAIN ANAL VOL 4, No 3, 1963.

③ Allison, I. M., : THE ELASTIC CONCENTRATION FACTORS IN SHOULDERED SHAFTS, PART III: SHAFTS SUBJECTED TO AXIAL LOAD, AERONAUT. Q. VOL. NO. 13, 1962.

④ RAYMOND J. ROARK, WARREN C. YOUNG: FORMULAS FOR STRESS AND STRAIN, Fifth Edition, McGraw-Hill Book Company, 1975.

STRESS CONCENTRATION "k"

$$k = K_1 + K_2\left(\frac{2h}{D}\right) + K_3\left(\frac{2h}{D}\right)^2 + K_4\left(\frac{2h}{D}\right)^3$$

$h = .156$
$r = .030$
$D = .622$

STRESS CONCENTRATION "K"

$$"K" = K_1 + K_2\left(\frac{2h}{D}\right) + K_3\left(\frac{2h}{D}\right)^2 + K_4\left(\frac{2h}{D}\right)^3$$

$$\frac{h}{r} = \frac{.156}{.030} = 5.2 \qquad \sqrt{\frac{h}{r}} = 2.280$$

$$\frac{2h}{D} = \frac{(2)(.156)}{.622} = .502 \qquad \left(\frac{2h}{D}\right)^2 = .252 \qquad \left(\frac{2h}{D}\right)^3 = .127$$

$$K_1 = 1.225 + 0.831\sqrt{\frac{h}{r}} - .010\,\frac{h}{r}$$
$$= 1.225 + (.831)(2.28) - 0.10\,(5.2)$$
$$= 1.225 + 1.895 - .052 = \underline{\underline{3.068}} = K_1$$

$$K_2 = -1.831 - .318\sqrt{\frac{h}{r}} - .049\ h/r$$
$$= -1.831 - (.318)(2.28) - .049\,(5.2$$
$$= -1.831 - .725 - .255 = \underline{\underline{-2.811}} = K_2$$

$$K_3 = 2.236 - .522\,(2.28) + .176\,(5.2)$$
$$2.236 - 1.190 + .915 = \underline{\underline{1.961}} = K_3$$

$$K_4 = -.630 + .009\sqrt{\frac{h}{r}} - .117\,\frac{h}{r}$$
$$= -.630 + .021 - .608 = -1.217 = K_4$$

$$2\frac{h}{D} = .502 \qquad \left(\frac{2h}{D}\right)^2 = .252 \qquad \left(\frac{2h}{D}\right)^3 = .125$$

$$K = K_1 + K_2\,(.502) + K_3\,(.252) + K_4\,(.127)$$
$$= 3.068 + (-2.811)(.502) + 1.961\,(.252) + (-1.217\,(.127)$$
$$= 3.068 - 1.411 + .494 - .155 = \underline{1.996}$$

STRESS CONCENTRATION FACTOR $K = 1.996$

 [redacted]

THE STRESS CONCENTRATION FACTOR 1.996 FOR BOLT (-.009) PART NO. ON DWG NO. V073-340120 - ROCKWELL INTERNATIONAL CORP. WILL BE RELATED TO THE PREVIOUS STRESSES AS CALCULATED.

THE TEST BOLT HAD A "D" OF .500" AND NOT .622" + THUS "h" would be .094 instead of .156. THIS IS A 40% REDUCTION

HOWEVER IN CALCULATING THE STRESSES IN THE G. A. S. PROGRAM THE WORST BOLT MUST BE ANALYZED FIRST.

FROM PAGE 10, IT WAS CALCULATED THAT THE STRESS IN THE CRITICAL SECTION WAS 222,500 #/□".

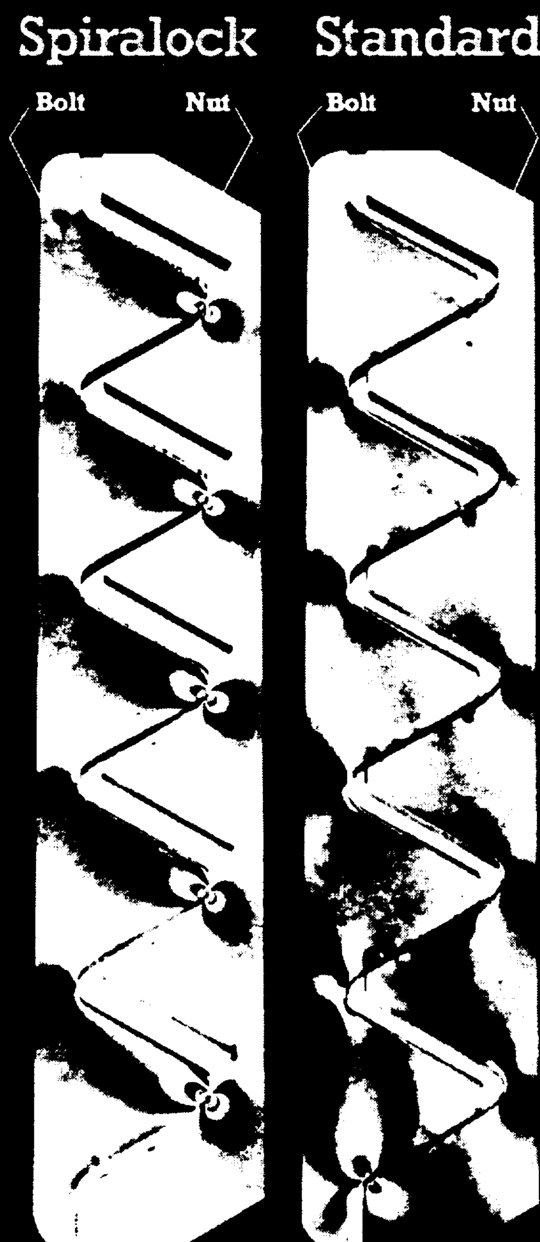
(222,500)(1.996) = <u>444,110</u> #/□". THIS IS A VERY HIGH CALCULATED STRESS AND EFFORTS SHOULD BE MADE TO SIMULATE THIS CONDITION BEFOR FINAL DESIGN.

TRUE, LOCAL YIELDING CAN OCCUR IN A STRUCTUR. AND IT DOES MANY TIMES. BUT THIS CASE IS A FORM OF YIELDING WHICH COULD RESULT IN AN IMMEDIATE CATOSTROPHIC FAILURE.

THIS CRITICAL STRESS IS FURTHER MORE CRITICAL BECAUSE IT IS APPLIED SUDDENLY AND IT DOES NOT GIVE THE INTERNAL CRYSTALS OF INCONEL TIME TO ORIENT THEMSELVES TO THE IMPACT.

FURTHER THE TOP OF THE BOLT COULD RECEIVE THE IMPACT AT ONE SIDE ONLY CAUSING A FURTHER DISRUPTION IN THE LOAD PATH THROUGH THE BOLT AS ILLUSTRATED ON PAGE 13.

FURTHER THE SLIGHTEST AMOUNT OF BENDING* IN THE BOLT OR THE <u>FRAME</u> COULD CAUSE ANOTHER STRESS CONCENTRATION FACTOR.
(* OR TWIST)



Photoelastic study dramatically illustrates the distribution of stress in the Spiralock thread design versus standard threads.

HOW SPIRALOCK DOES IT.

This photoelastic study shows just how Spiralock compares with standard thread designs. The single contact point at the root of each Spiralock thread can be clearly seen. Stress is evenly distributed throughout the nut. So is locking power.

There's nothing quite equal to Spiralock.

It offers the best available solution to all three of the major nut/bolt failure problems: overstress, fatigue, and vibration.

Standard threads concentrate the load at the first engaged thread. Spiralock eliminates that single, intense, stress point—and with it, most bolt failures.

In corrosive applications, both Spiralock locknuts and bolts can be protectively plated—and still be highly reusable. The free spinning design lets the nut run on and off without galling or wiping off the plating.

Need extra locking safety?

Our crimped Spiralock (upper portion of thread is geometrically reformed) adds another dimension of locking power. It's a belt-and-suspenders safety margin for your most critical applications.

KAYNAR

FIG. 7

DESCRIPTION OF NUT THREADS

Details of Photoelastic study on reverse side

Figure 7. Description of Nut Threads

THE USE AND MISUSE OF SIX BILLION BOLTS PER YEAR

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Engineering Services Division
NASA/Goddard Space Flight Center

Abstract: Official published reports of major structural failures (such as the Kansas City Kemper Arena) have identified the cause of many of these failures to be the time-related failure of bolts. It is often difficult to identify the precise root cause of time-related structural failures such as: (1) fracture of high-strength/low-yield bolts due to repeated high loads, (2) bending caused by yielding due to many overloads, and (3) loosening of nuts due to low-frequency vibration (even though loads may be low). Modern design trends such as using high-strength/low-yield bolts, torquing into the yield region, and improper lubrication practices could lead to many types of time-related failures. The writer believes that time-related failures are strictly an engineering problem that could be avoided with good design. If the designer has a clear concept of bolt analysis, bolt problems can be avoided by the use of existing analytical and testing methods. This paper hopes to explain why erroneous ideas have persistently plagued bolt analysis. Design concepts presented will lead to a more thorough understanding of the bolt problem.

Key words: Bolts; nuts; washers; torquing; lubrication; vibration.

Introduction: Some current trends in engineering design and application of bolted structures could lead to an increase in the number of time-related failures of bolts and nuts. Some of these trends are: (1) torquing into the plastic region, (2) inadequate knowledge of the friction factor and the use thereof, (3) neglecting to consider the bolt problem in early design, (4) inadequate knowledge of the types of loads caused by shock transients, (5) not testing bolted structures that are difficult to analyze and, in some cases, performing the wrong tests, and (6) relying too much on computer analysis without proper engineering judgment. Because space does not permit adequate coverage of this material, this paper will: (1) present several design models describing the deflection of bolts; (2) describe these models in mathematical terms; (3) from these mathematical models, form *design* tools and illustrate them graphically; and (4) provide a number of design suggestions and hints for minimizing the possibility of a time-related bolt failure. Time-related failures can best be prevented on the drafting board, not after the trouble is discovered. Therefore, this paper is written primarily for the designer.

A discussion of torquing and the effects thereof can best be visualized in terms of the deflections of a bolt to which torque is applied, the nut that anchors the bolt, and the frame structure that is secured by the bolt/nut combination. In practice, the undeformed bolt is installed in an undeformed frame and the nut is turned up; the bolt extends and the frame compresses. It is difficult to visualize the types of deflections of both the bolt and the nut while visualizing the deflection of the nut turning up. To aid this visualization, a model is made (Figure 1) that will illustrate these deflections by turning the nut up to its final position outside the frame. The bolt is then extended outside the frame. The frame is compressed before the extended bolt is slid into the compressed frame. By mentally separating these actions, one can more readily visualize the resulting deflections.

Figure 2 illustrates this same model. First, the nut is turned up to the position it would be in if it were torqued up. Second, the bolt is extended to the final position it would take if it were torqued up. Third, the frame is compressed to the final position it would take if it were torqued up. Then the extended bolt is slid into the compressed frame.

Figure 3 shows the same extensions and compressions as those in Figure 2, but the effects of torque are depicted for a stiff frame and a thin (or less stiff) frame. Because it is more flexible than a stiff frame, the thin frame shows larger deflections. Through the following examples, the bolt stiffness is kept constant, whereas the frame varies from thin to stiff. Figure 4 is a graph of the loads and deflections for the bolt and normal frame. The deflection (Δ) is a function of the force (P), the length (L), the area (A), and the modulus of elasticity (E).

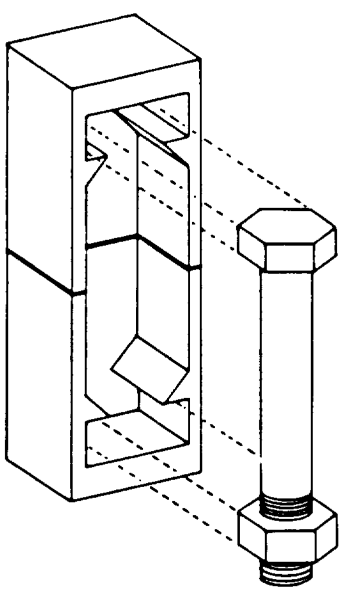


Figure 1. Bolt and Frame

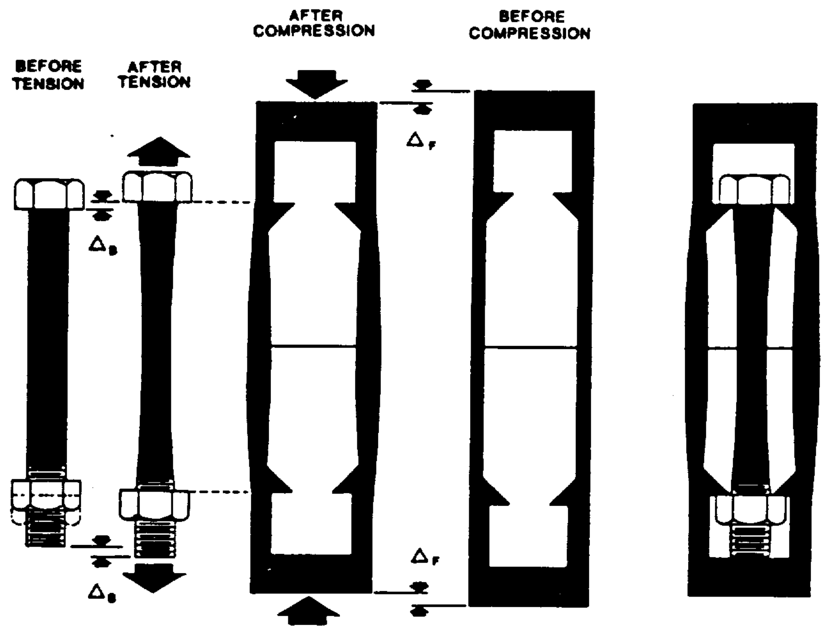


Figure 2. Deflection of Bolt and Frame Due to Torquing

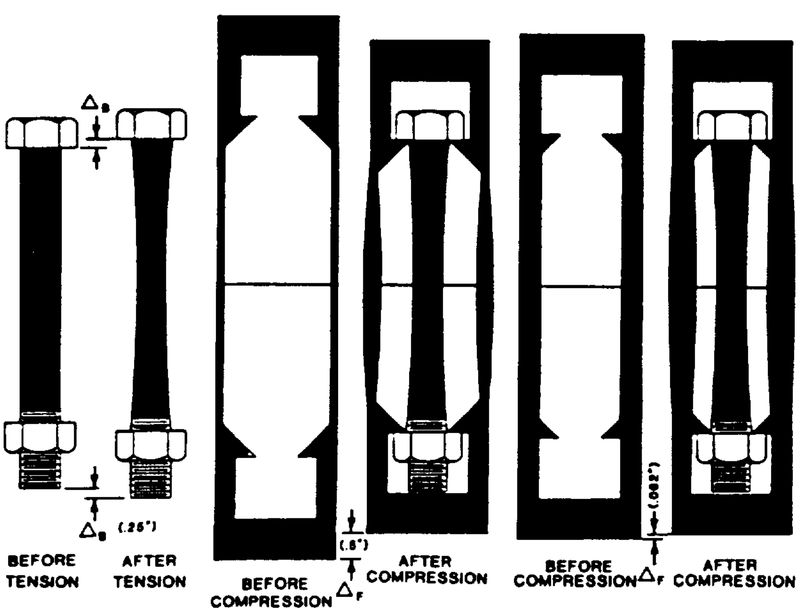


Figure 3. Deflection of a Thin and a Stiff Frame Due to Torquing

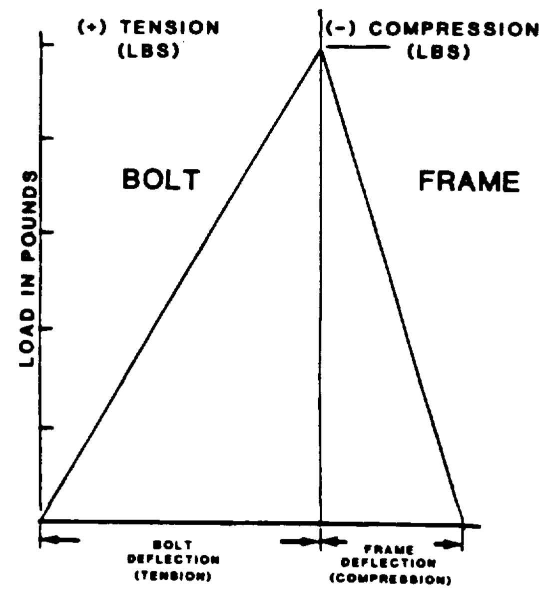


Figure 4. Loads and Deflections on Bolt and Frame

For example, assume for the bolt that P_B = 50,000 lb, L_B = 10 in., A_B = 0.1 in^2, and E_B = 10,000 lb/in^2. Under such conditions, a bold elongation of 0.5 in. will result as shown by:

$$2\Delta_B = \frac{PL}{AE} = \frac{(50{,}000)\,(10)}{(0.1)\,(10{,}000{,}000)} = 0.5 \text{ in.}$$

$2\Delta_B$ is plotted as the curve on the left in Figure 4. The only difference between the curves used in the study of bolts and nuts and other load deflection curves is that the frame deflection curve is reversed as shown in Figure 4. For example, assume for the frame that P_F = 50,000 lb, L_F = 10 in., A_F = 0.2 in^2, E_F = 10,000,000 lb/in^2. Such frame parameters will result in a frame compression of 0.25 in. as shown by:

$$2\Delta_F = \frac{PL}{AE} = \frac{(50{,}000)\,(10)}{(0.2)\,(10{,}000{,}000)} = 0.25 \text{ in.}$$

The 50,000-lb compression load in the frame is equal to the 50,000-lb tension load in the bolt. Figure 5 shows a frame before torque, after torque, and with an external tension load after pretorque. Note that the deflection of the frame (Δ_F) is caused by the torquing, followed by a *reversal* in deflection (Δ_E) when the external tension load is applied. Part of the bulge in the frame is lost; the frame load goes down. When the external tension load is added, the bolt continues to neck down even more as the bolt tension load increases. The belief that the bolt load remains the same until preload is overcome is a common error.

The external tension load lowers the frame load and raises the bolt load. Any motion in the frame causes the bolt to move with it, even though the stress is relieved in the frame and increased in the bolt. Figure 6 shows

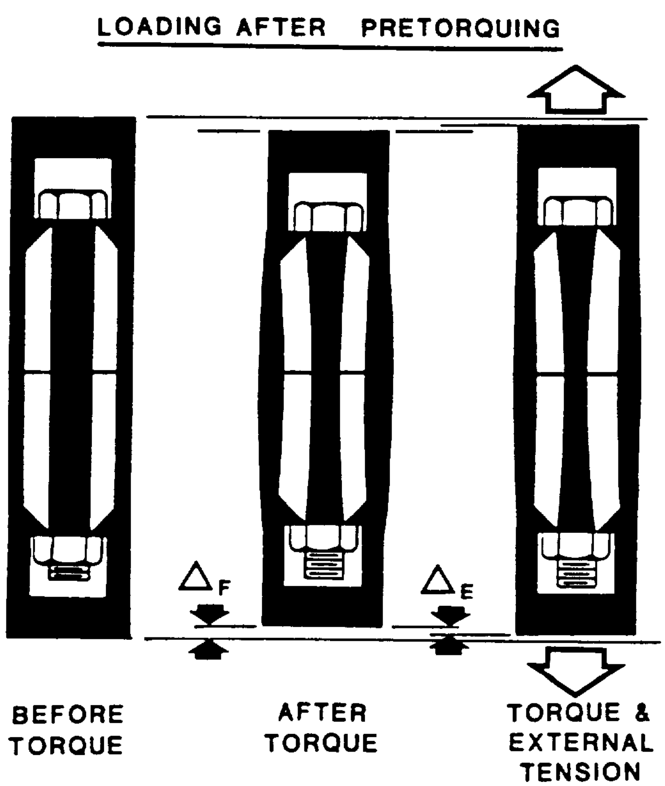


Figure 5. Deflections After Torquing and External Load of 30,000 Pounds

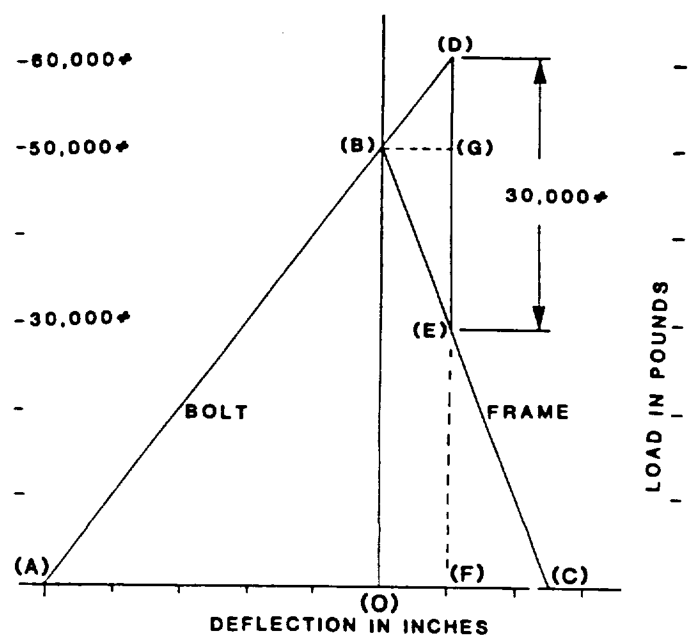


Figure 6. 30,000-Pound Tensile Load Added to 50,000-Pound Pretorque

(3)

the graphical method for representing these forces and deflections. The preload is 50,000 lb and the external tension load is 30,000 lb. As the bolt is extended in pretorque along AB, additional tension loads after pretorquing must follow the same slope to D. The frame is compressed along CB, and backs off to E when the external tension load after pretorque is applied. The additional bolt load is GD, which is measured along the vertical axis. The load lost to the frame is EG. GD and EG must add up to the external load of 30,000 lb.

The deflection of both the bolt and the frame ($2\Delta_E$), or GB, represents the additional extension of the bolt and extension of the frame, *after* the frame was previously pretorqued in compression. Thus, the extension in the frame is simply a loss of compression that was already there. The total extension of the bolt is AO + OF. The compression of the frame after torquing is CO. By the external load, the frame is relieved of some of that compression (OF), with a final deflection (in compression) of CF. In the case of an external compression load (to be discussed later), the process is simply reversed.

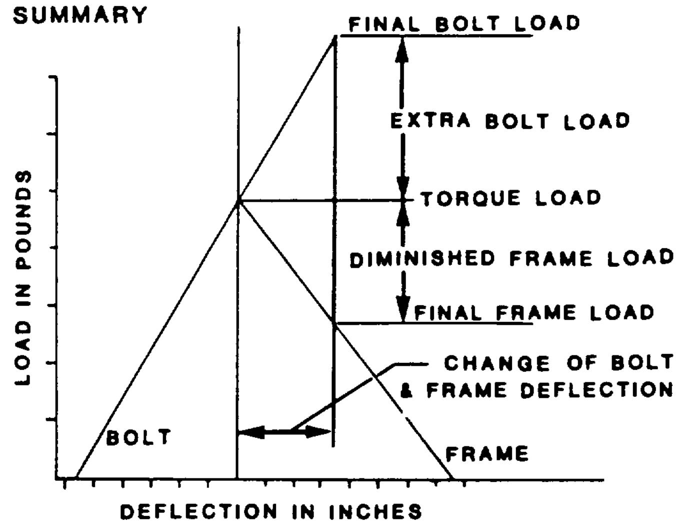
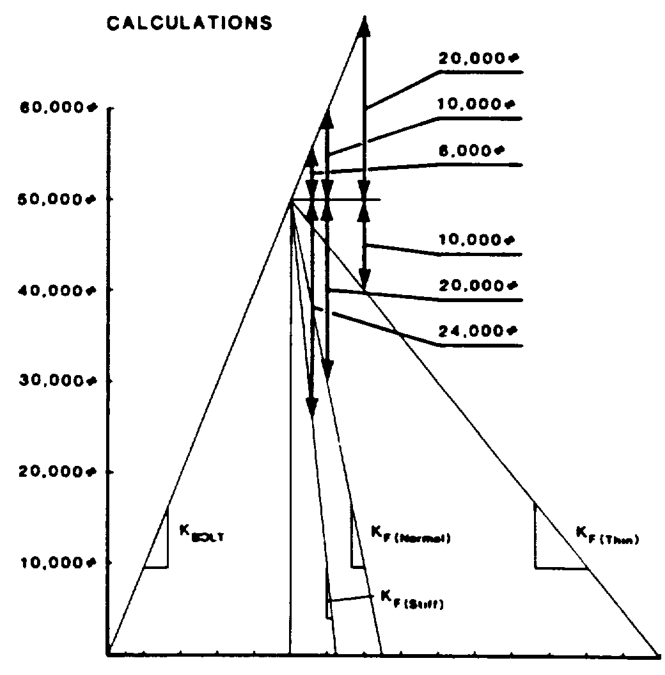


Figure 7. Summary of Loads and Deflections

Figure 7 summarizes all previous terms. The extra bolt load and the loss of frame load are obvious. The change in bolt and frame deflection shows that the bolt is extended farther but that the frame compression is reduced.

Figure 8 superimposes the load deflection curves of three frames of varying stiffness (i.e., stiff, normal, thin) while holding the bolt stiffness constant. Such a presentation can be used as a design tool to show the change in bolt and frame loads by changing the stiffness of the frame. As previously described, there are eight variables: area, length, force, and modulus of the bolt and of the frame. The method is to change one variable at a time and then relate the other variables to that change. In Figure 8, the four variables of the bolt are held constant, along with the length, force, and modulus of the frame. The frame area is the only variable.



$K_B = \frac{50,000\#}{.50\ IN.} = 100,000\ \#/IN.$ $\quad K_{F(Normal)} = \frac{50,000\ \#}{.25\ IN.} = 200,000\ \#/IN.$

$K_{F(Stiff)} = \frac{50,000\#}{.125\ IN.} = 400,000\ \#/IN.$ $\quad K_{F(Thin)} = \frac{50,000\#}{1.0\ IN.} = 50,000\ \#/IN.$

Figure 8. Loads and Deflections
(normal thin and stiff frame)

For example, assume the preload to be 50,000 lb as previously plotted. The deflection of the bolt is 0.5 in. and the deflection of the normal frame is 0.25 in. due to the preload. When an external tension load of 30,000 lb is added, the bolt load is

increased 10,000 lb and the normal frame load is decreased 20,000 lb because the normal frame stiffness (K_F) is twice the bolt stiffness (K_B): K_F = 200,000 lb/in. and K_B = 100,000 lb/in.

The design tool can now be used as follows. Bracket the proposed (or normal frame) design by selecting a thin and stiff frame at random. This is achieved by selecting lower and higher frame areas that change K_F. For the thin frame, cut the area of the frame to one-half that of the bolt. Then K_B = 100,000 lb/in and K_F = 50,000 lb/in. This is plotted on Figure 8 as $K_{F(thin)}$. Note that softening the frame makes the change in bolt load due to external load go from 10,000 to 20,000 lb. For the stiff frame, increase the area of the frame by 4, which raises $K_{F(stiff)}$ to 400,000 lb/in (Figure 8). The change in bolt load due to external load goes down to 6,000 lb. Simply changing the area of the frame can control the loads on the bolt and frame. Changing the frame from steel to aluminum lowers the stiffness by one third. Plastic frames can lower the stiffness to 1/15th of the original stiffness or even lower. The bolt load goes up very fast. The bolt parameters can also be changed. The bolt can be necked down or made longer to decrease the stiffness, or it can be made larger or with a higher modulus.

Figure 9 represents a bolt pretorqued up to the yield point. Subsequent external tension loads after pretorquing cause the bolt to yield. The bolt loses some of its pretorque as illustrated. In time-related failures, this practice could be dangerous as the load is repeated. From a design point of view, it is important to select the proper stiffness for the bolt and the frame to ensure long life. Increasing the strength of the bolt does not always solve the problem. The stiff bolt has less yield, which makes it brittle. Selecting a weaker bolt with a low yield may not solve the problem because constant yielding lowers the clamping force and allows the bolt to bend and fatigue.

Figure 10 graphically represents a bolt that has been pretorqued and then subjected to an external compression load instead of a tension load. The frame previously in compression due to pretorque must take more compression and thus the frame load goes up. Because the bolt must move with the frame, it loses load since it was previously in tension, and the new external load is compression. The compression design considerations for time-related failures are:

1. In the case of a stiff frame, the loading within the frame can increase rapidly with external load.

2. Excessive additional compressive load will relieve the preload in the bolt and cause the nut to back off, or the frame could lose its clamping force. The frame would lose its resistance to shear, and the bolt will bend, which could lead to early fatigue. The impact load of gapping could be quite serious.

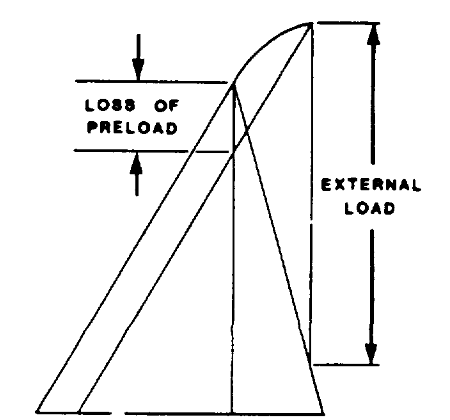


Figure 9. Bolt Plastic Tension

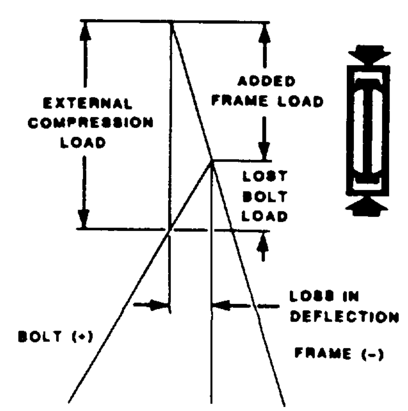


Figure 10. External Compression

3. With a plastic or thin frame, $K_{F(thin)}$, the load in the bolt will drop off *very fast*, and, in some designs, it is difficult to maintain a consistent clamping force through the years.

Figure 11 graphically represents a vibration load on a bolt/frame combination. The small figure on the upper right demonstrates the first cycle of vibration. The bolt/frame combination is first subjected to a half cycle of external compression and then a half cycle of external tension. The central sketch represents the tension and compression forces acting on the bolt and frame exactly as the tension and compression on the bolt and frame previously discussed. The small sine curve in the lower center represents the deflection, in which the motion goes from compression to tension. With proper pretorque, they are as good as glued together. On the left sketch, the bolt vibration load and the frame vibration load are plotted against time.

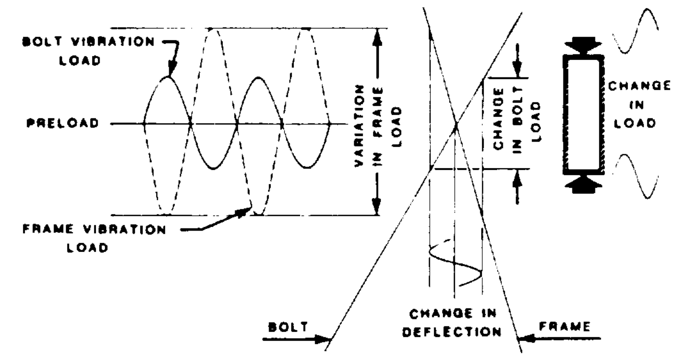


Figure 11. Sine Vibration Load and Deflection

The vibration is represented as a sine function to show simple harmonic motion. Other vibration loads will be discussed later. Note that the frame goes through a greater change in load than the bolt and is more subject to fatigue. If the frame is stiffer than the bolt, more force is required to move the frame 1 in. than to move the bolt. By approaching an infinite frame stiffness, it is possible to keep the load reversal of the bolt to a minimum. If the frame is not as stiff as the bolt (such as a plastic frame), the bolt will have a severe fatigue problem. It is immediately obvious that loads such as those illustrated here can cause severe fatigue problems in both the bolt and the frame. If the frame is more susceptible to fatigue, it is better to make it softer than the bolt or simply increase the size of the bolt. Again, the torque reduction due to bolt load loss can be a difficult design problem. Many of these problems can be controlled by carefully selecting the pretorque. Too much torquing will throw the bolt into the plastic region and bring about an early fatigue failure. Too little torquing can cause gapping.

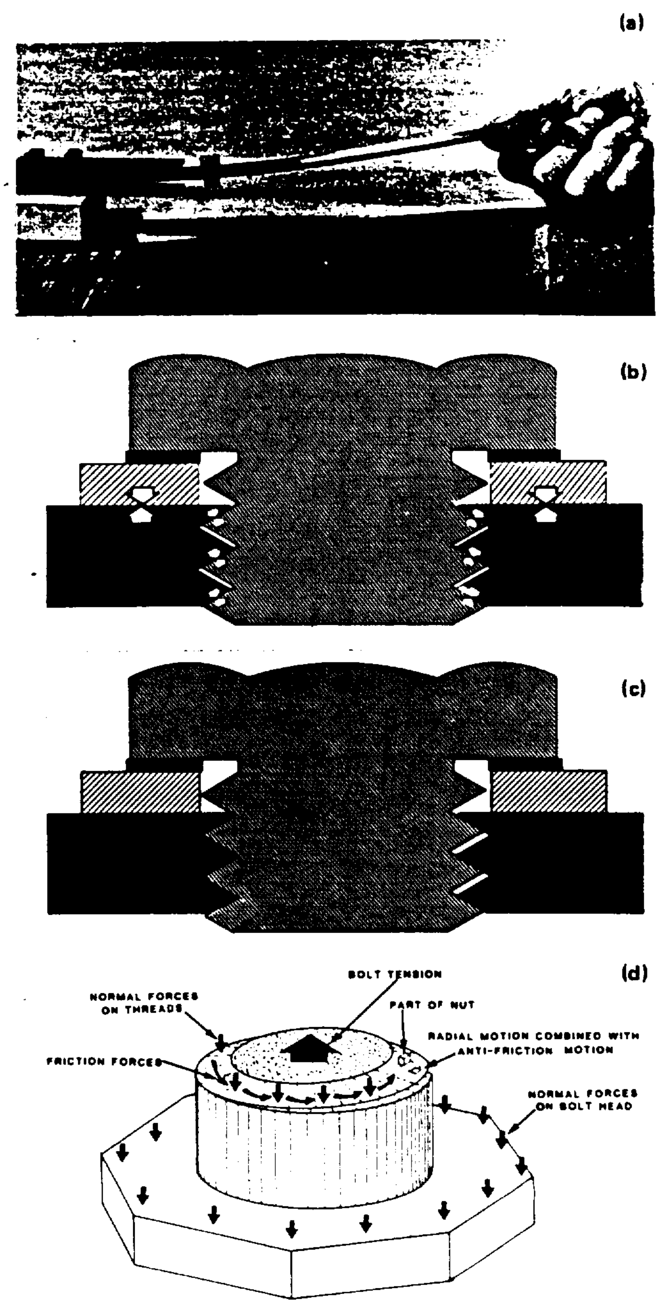


Figure 12. Loss of Torque (inverted bolt section)

Figure 12 is a step-by-step illustration of the Haviland theory* of bolt loosening due to a reversal of loads. In

*"A Logical Approach to Secure Bolting and Locking," Girard S. Haviland, Loctite Corporation, Newington, Connecticut, 06111.

[redacted]

Figure 12a, two thin plates are bolted together with a single bolt. This bolt is tightened to keep the plates from shearing, one against the other. Figure 12b shows the forces existing before bending. The torquing forces holding the plates together are shown, as well as the forces on the threads causing the clamping force. As the two thin plates are bent back and forth together (Figure 12a), the bolt tends to slide to the left as shown in Figure 12c. As the nut thread slides sideways, it takes the course of least resistance. In Figure 12d, the radial motion is accompanied by a motion downhill against the friction holding it in place, friction being the only thing holding the pretorque. When the motion is sideways, pretorquing is simultaneously lost. The motion is graphically demonstrated by the motion of "part of nut" in Figure 12d. The results of vibration loads from a transverse shock and vibration machine are reprinted from a report by Haviland (Figure 13). This type of loading is violent because the initial impact breaks the friction of the nut while the side motion of the threads steadily pushes the nut downhill, causing loss of the pretorque. Figure 13 shows time-related failures with different types of thread lockers.

Figure 14 depicts a vibration test setup for a sounding rocket with a 2-g sine input at the NASA/Goddard Space Flight Center (GSFC). Many responses of the panels, struts, and bolts were recorded during the test program. Figure 15 shows a shock transient followed by a high-frequency harmonic and the fundamental frequency of the structure. Figure 16 is the sine response with a light fourth harmonic. Even though one frequency is driving the base of the sounding rocket, the response (caused by the resonant response of many of the panels, etc.) can cause motions much higher than the original input. Note, however, that the fundamental frequency is always present in all four figures, along with the other harmonics. In similar fashion, Figure 17 shows the fundamental frequency along with a strong third harmonic. Figure 18 shows the fundamental frequency with a second harmonic and other higher harmonics. The higher harmonics tend to break the friction, but the fundamental frequency moves the nut to one side and loosens up the assembly. The motion actively causes a loss of initial pretorque.

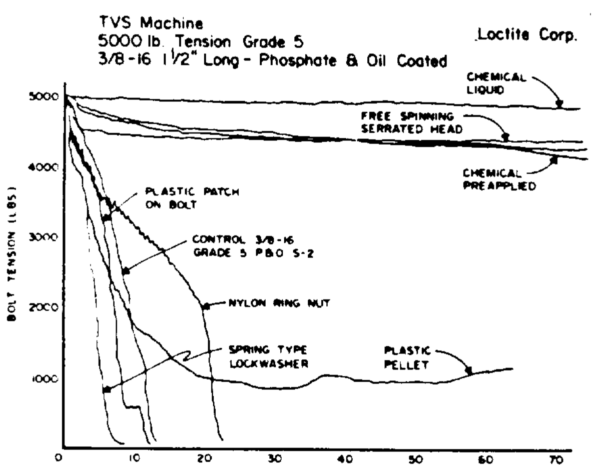


Figure 13. Transverse Shock and Vibration Machine

Figure 14. Vibration Test

Figure 15. Shock Transient Vibration Response

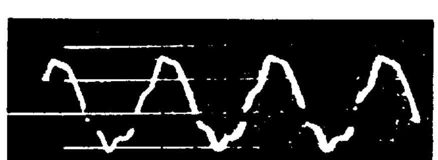


Figure 16. Fundamental and 4th Harmonic

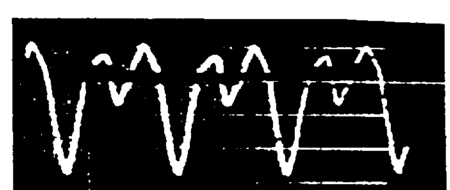


Figure 17. Fundamental and Strong 3rd Harmonic

The Haviland theory is important in the study of shock and vibration on bolts and frames. Figures 15 through 18 typify structures undergoing vibration tests. Not realized is the presence of three-dimensional motion. It is almost impossible to obtain the high frequencies without cross-talk causing secondary motions in the other two planes. This means that: (1) no matter where the bolts are, they will receive vibration if vibration is present on the structure; and (2) if low frequency causes the bolts to break away and back off, high frequencies can cause subharmonics that may be low in nature but nevertheless sufficient to cause the bolts to back off.

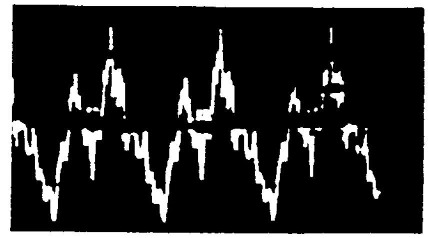


Figure 18. Fundamental 2nd and Higher Harmonics

Tests were performed at GSFC to verify the Haviland theory. When the structure was vibrated at the low-frequency, high-amplitude loads (5 g), the nuts backed off in a matter of seconds. When 20 g was applied at 1000 Hz, the bolts had not moved after 10 minutes. In similar fashion, to simulate the multifrequencies of vibration that can occur, random vibration was applied. At 10 grms from 20 to 400 Hz, the nut separated in seconds. At 10 grms from 200 to 2000 Hz, the bolts did not move in 10 minutes. These tests substantiate the theory that low-frequency excitations cause a bolt to back off (Haviland, ibid).

The significant motion in the bolt that involved the $K_{F(thin)}$ frame was previously discussed. Plastic reinforced with fiber is a good example. This significant bolt fluctuation of load could literally tear the frame apart if the clamping force were diminished. Some of these problems were illustrated by taking the frame of Figure 14 and letting it vibrate until some of the bolts back off (Figure 19). Note that gapping caused damage to the plastic surface before the bolts backed off. Figure 20 shows the result of the loosening of the bolts, which permits the frame and bolt to bend and wear into the plastic frame. Even if the nut does not back off, this wear on the contact surface can cause damage. If the pretorque is too high, the additional load will cause it to yield immediately. If the torque is too low, gapping will result and damage the surface. With plastics, the pretorque must be carefully calculated and controlled.

Figure 21 shows the result of loose bolts in the frame acting as a lever and stress concentration to wedge open the plastic frame and destroy it. These types of failures can occur frequently in thin frames. Further, the loose bolt could bend and break. A good threadlocker will help to maintain the pretorque. However, even a good thread-locker cannot prevent excessive wear if the bolts are torqued too high or too low. Plastic frames are subject to large deflections during vibration. High torques will immediately cause the plastic surface to yield, and low torques will cause gapping even though the nut does not back off. All thread-lockers do not act like those illustrated by Figure 13, but all vibrations are not as violent as the shock transients produced by the transverse shock and vibration machine. Furthermore, a particular type of tread-locker may not work under all types of vibration and shock.

Figure 19. Frame Yield Allows Bolts to Back Off and Gap

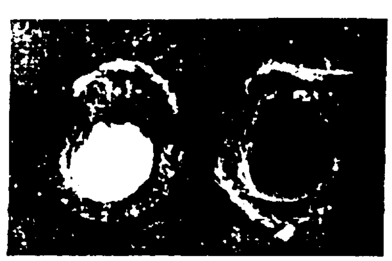


Figure 20. Frame Yield and Gapping Cause Wear in Frame

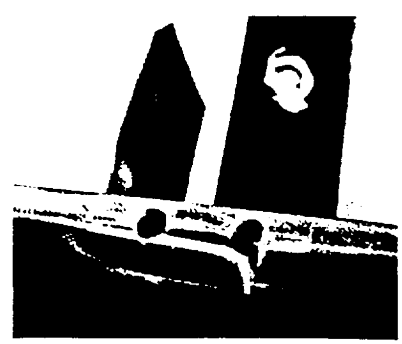


Figure 21. Loose Bolts Bend and Break Frame

There are many types of shocks and vibrations acting in structures. These loads can cause many different types of reactions in bolts and nuts and, in some cases, can cause them to back off. Haviland gives the strongest clue to date as to the cause of much of this trouble. When trouble occurs, experience has clearly demonstrated that a good test of the assembly is essential.

Figure 22 is another example of a fabricated thin frame rather than two solid plates bolted together. This frame, a model simulation of a structural attachment point on the space shuttle, was used to evaluate payload mounting-bolt loads.

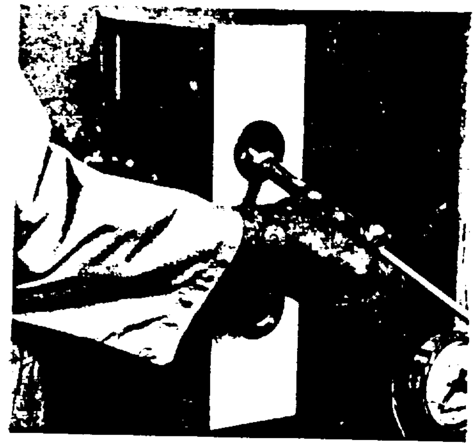


Figure 22. Thin Frame

Design Suggestions: Figure 23 shows the lubrication of the threads and under the head of the bold. There is little or no control with bolts that are not lubricated. If a thread-locker is used on the threads, the bolt must be lubricated under the head. Although thin oil can be used, there may be none left 2 years later. When possible, an *automatically controlled torque wrench* should be used. Many of these are on the market today. If an application is critical, a technique illustrated in Figures 24 through 27 can be used. First, the torque is calculated for bringing the assembly to yield. It is then torqued up in five or ten equal increments of torque and marked each time. Figure 24 illustrates the marking of 300 ft-lb of torque. After this torque was applied, it was found that a torque of 325 ft-lb would yield some part of the assembly (Figure 25). This significant yield is never necessary but is used here to illustrate the process. As soon as yield is noticed, back off the torque to approximately 65 percent of the yield value. Although not all materials are the same, this percentage is a safe torque. Figures 26 and 27 show the torquing of a 1/4-20 bolt. Torques of 10 through 70 in-lb were marked as the torquing was applied (Figure 26). When 80 in-lb were applied, however, something yielded. It was necessary to back off to 65 percent. There is no reason why an automatic torque wrench cannot be used to torque to yield and then be backed off. GSFC has been careful to follow these processes and has used strain-measuring gages to hold torques to ±10% accuracy.

Figure 23. Lubrication

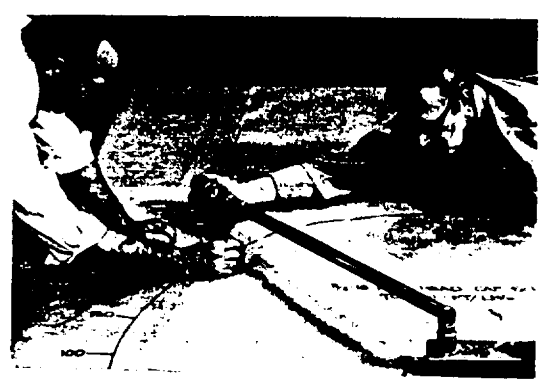


Figure 24. Torque of 3/4-10 Socket Head Capscrew

Figure 28 is a photostress model showing the severe stress concentrations caused by a sharp radius under the head and bending due to loss of torque. In time-related failures, it is desirable to use materials with large elongations and with generous radii in the threads and under the head of the bolt. Yield under the head can cause a loss in torque, with subsequent bolt bending and failure in time, and can further damage the frame in a time-related condition.

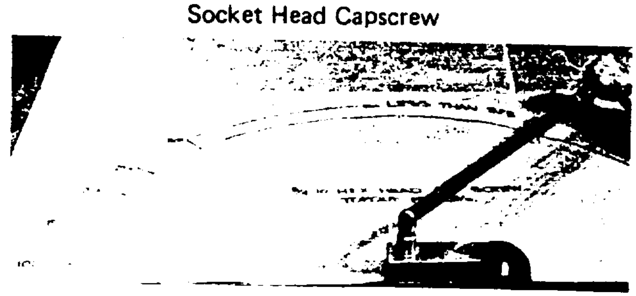


Figure 25. Yield

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Figure 26. Torque of 1/4-20 Socket Head Capscrew

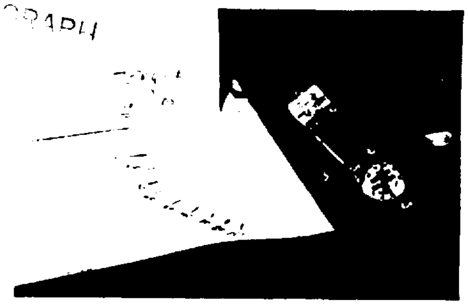

Figure 27. Yield

Figure 28. Stress Concentration Due to Bending

Figure 29 pictures the difference of stiffness in a frame with and without washers. The conical area of the frame contributes to the stiffness in K_F. K_F on the right is higher than K_F on the left.

Figure 30 demonstrates the necessity of having a strong clamping force (top and bottom) when shearing forces are acting on the frame (left and right). If the clamping force is inadequate, the shearing force will make the bolt bend as illustrated and force an early time-related failure.

Figure 31 shows that a cotter pin or lock wire will be of little use in preventing a nut from backing off. Less than half a turn usually loosens a nut.

In Figure 32, strong stiff washers are added to decrease the K_B of the bolt. There is more area under the washers than in the bolt. Because lowering K_B has the same effect as raising K_F, it is necessary to keep the bolt load low, particularly in low modulus materials such as plastics.

Figure 33 illustrates the desirability of using thread lockers, which prevent the bolt from moving sideways and thereby lock the bolt or nut into position. Lock washers have a tendency to dig into the surface and nick it, which prevents a good clamping action for the next use. Thread despoilers have the same effect. If they are too strong, they can cause the bolt to jam and lead to a considerable maintenance problem. They can also result in loose pieces of hard steel in the mechanism.

Figure 34 shows that it is possible to keep deflections low in vibration. Clamping the assembly on both ends results in a higher resonant frequency and lower deflections. According to the Haviland theory, large shearing deflections at a low frequency loosen bolts. High frequencies have low amplitudes.

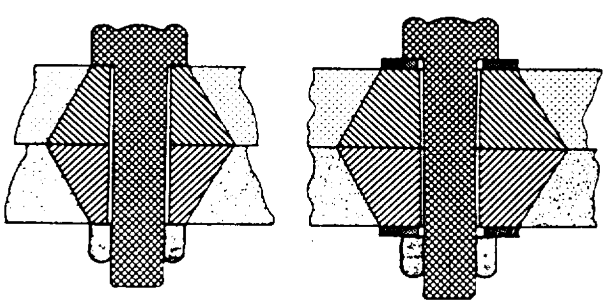

Figure 29. With and Without Washer

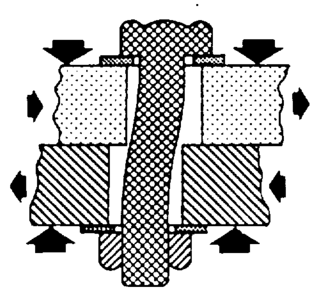

Figure 30. Clamping Force Prevents Shearing and Bolt Bending

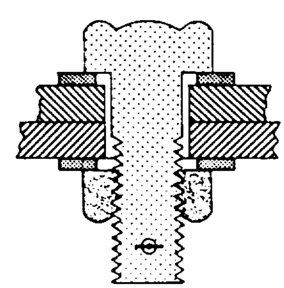

Figure 31. Cotter Pin and Lock Wire Do Not Prevent Loss of Torque

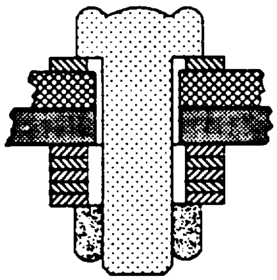

Figure 32. Long Bolt Decreases K_b and Absorbs Shock Transients

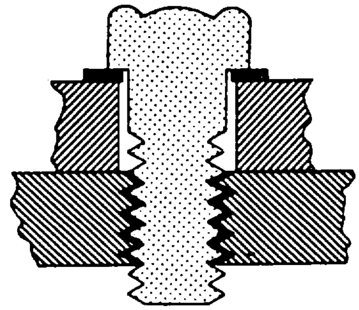

Figure 33. Use Threadlocker

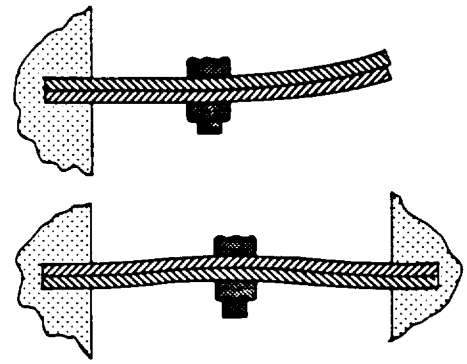

Figure 34. Keep Natural Frequency High

Figure 35 illustrates the effect of nonparallel alignment of both the bolt and the frame, which is detrimental to fatigue life. In the upper left, the frame surfaces are uneven. In the upper right, a hole was drilled at an angle. In the lower left, material is embedded under the bolt, and in the lower right, there is a protrusion under the nut. Fatigue life is considerably shortened if the bolt is bent during load reversals.

Figure 36 shows a typical bolt and nut under tension. The triangle in the left illustration represents the load or stress (through the threads). The first or second thread of the bolt and the first or second thread of the nut have far greater loads than the other threads. If the first nut thread breaks (lower right), the next thread will pick up the load. The bolt usually breaks from tension near the first thread as illustrated in the upper right because there is nothing to take up the load. If the nut should be designed with a large yield, it can take the brunt of the shock loads and relieve the stress on the bolt. This problem is another example of the importance of considering total design rather than selecting a bolt after the design is finished.

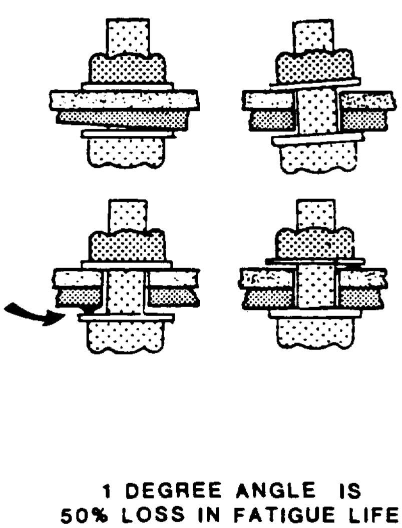

Figure 35. Misalignments Causing Bolt Bending

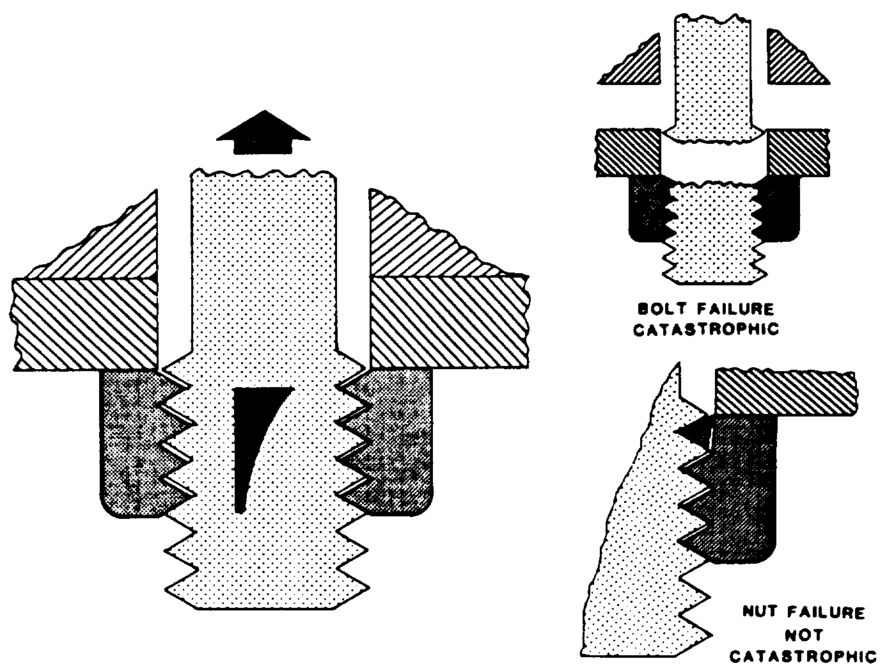

Figure 36. Bolt and Nut Thread Failure

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